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Inoue et al.

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(54) **CONTROLLER FOR SUPERCHARGER-EQUIPPED INTERNAL COMBUSTION ENGINE**

(52) **U.S. Cl.**
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(71) Applicants: **Mitsubishi Electric Corporation**, Tokyo (JP); **MAZDA MOTOR CORPORATION**, Aki-gun, Hiroshima (JP)

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(72) Inventors: **Junichi Inoue**, Tokyo (JP); **Takahiko Ono**, Hyogo (JP); **Toshiaki Maruo**, Hyogo (JP); **Takafumi Nishio**, Hiroshima (JP); **Yugou Sunagare**, Hiroshima (JP)

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Primary Examiner — Thai Ba Trieu

(74) *Attorney, Agent, or Firm* — Sughrue Mion, PLLC; Richard C. Turner

(73) Assignees: **Mitsubishi Electric Corporation**, Tokyo (JP); **MAZDA MOTOR CORPORATION**, Hiroshima (JP)

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(51) **Int. Cl.**

F02D 23/00 (2006.01)

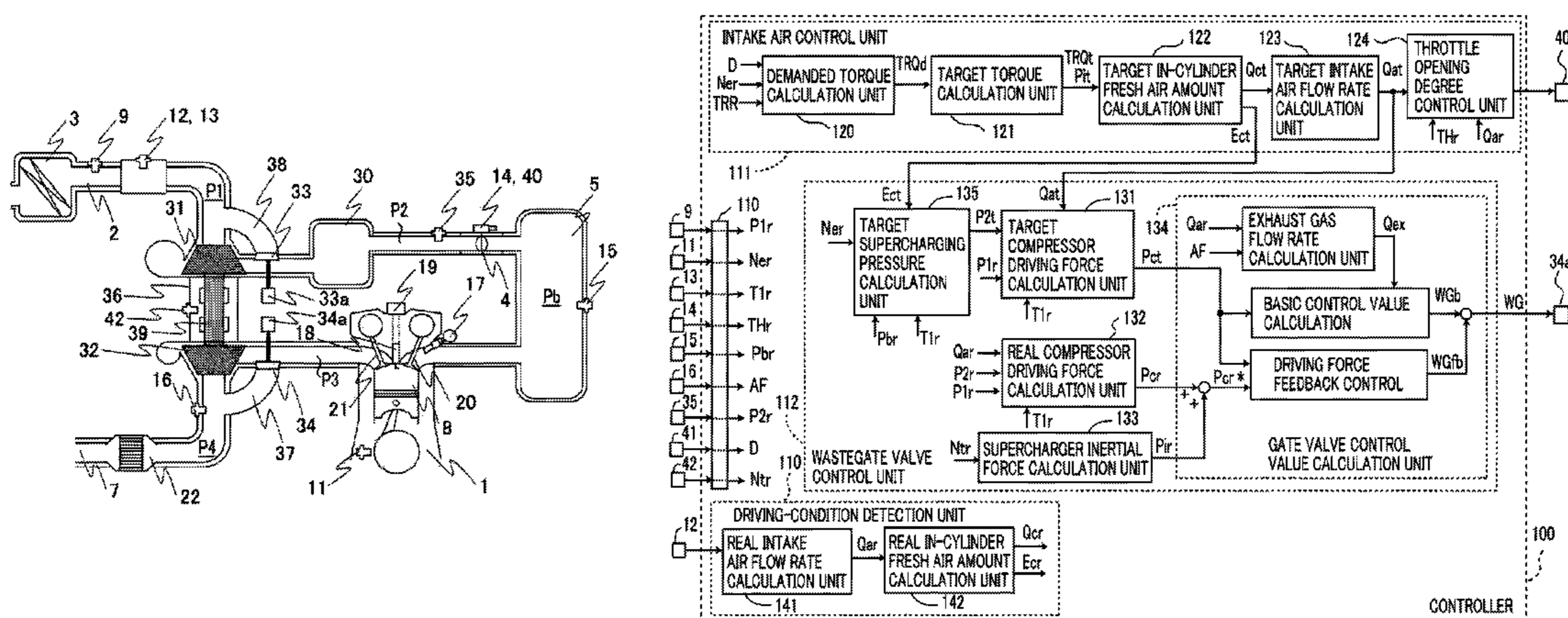
F02B 33/44 (2006.01)

(Continued)

(57) **ABSTRACT**

A controller, for a supercharger-equipped internal combustion engine, that can improve the feedback response of compressor driving force is provided. In a controller, inertial force produced by an inertial moment of a supercharger is calculated, based on a real rotation speed of the supercharger; then, driving force feedback control is implemented in which a gate valve control value, which is a control value for a gate valve actuator, is changed so that an addition value obtained by adding the inertial force to the real compressor driving force approaches a target compressor driving force.

7 Claims, 10 Drawing Sheets



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F02D 41/14 (2006.01)
F02D 11/10 (2006.01)
F02D 41/18 (2006.01)

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(2013.01); *F02D 2041/1409* (2013.01); *F02D*
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(2013.01)

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Y02T 10/42
USPC 60/602, 605.1, 611; 701/102–104
See application file for complete search history.

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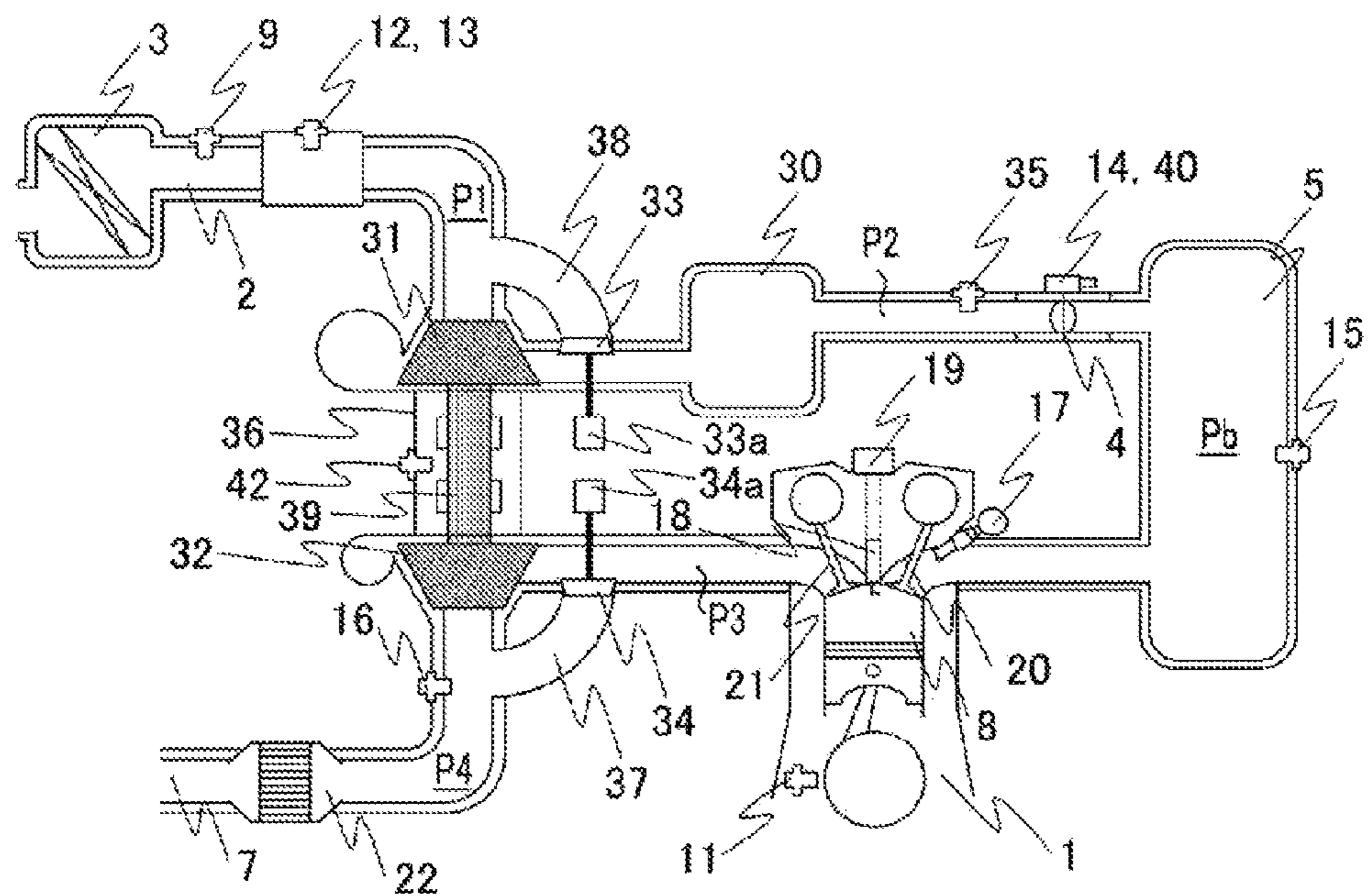
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FIG. 1



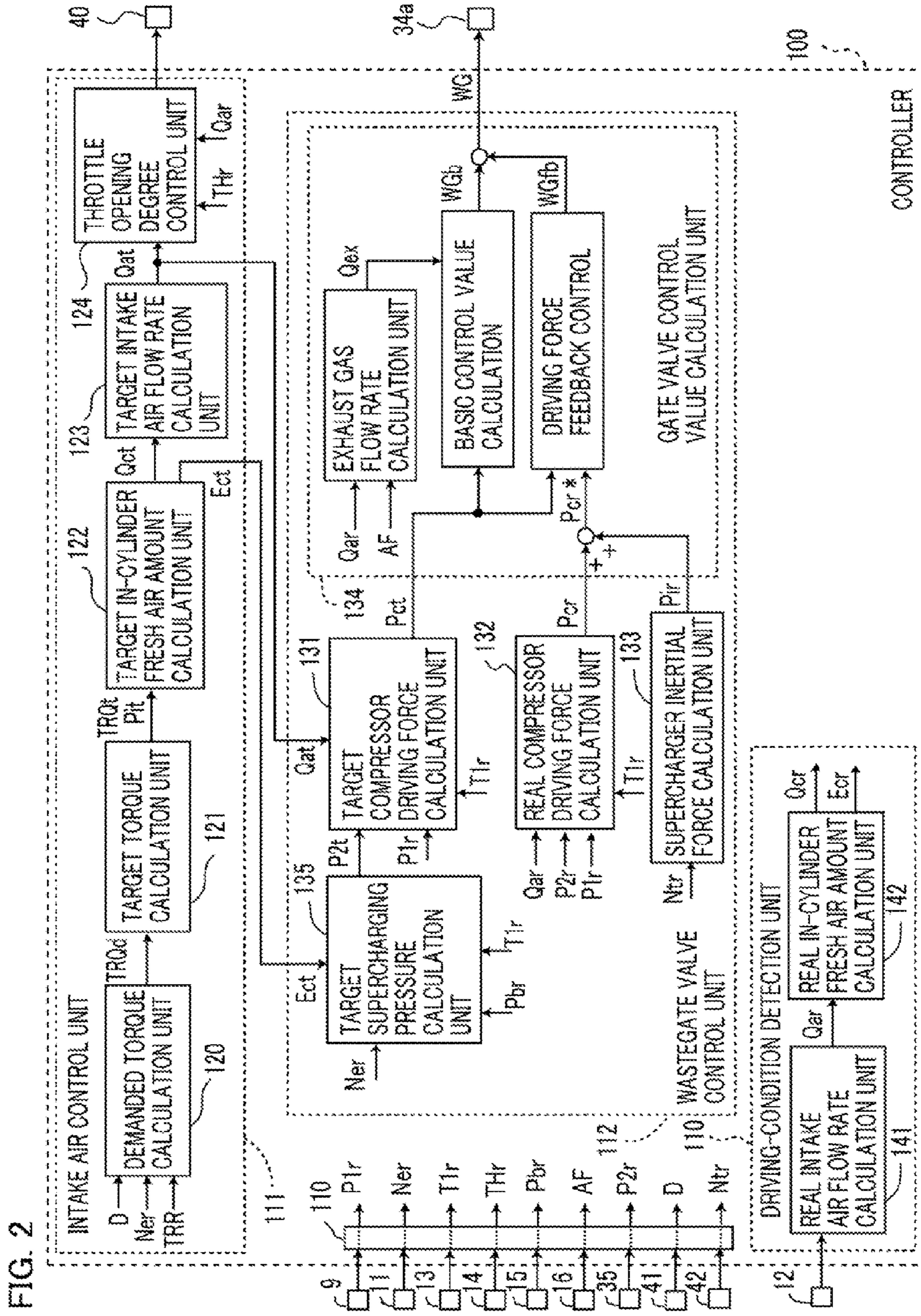


FIG. 3

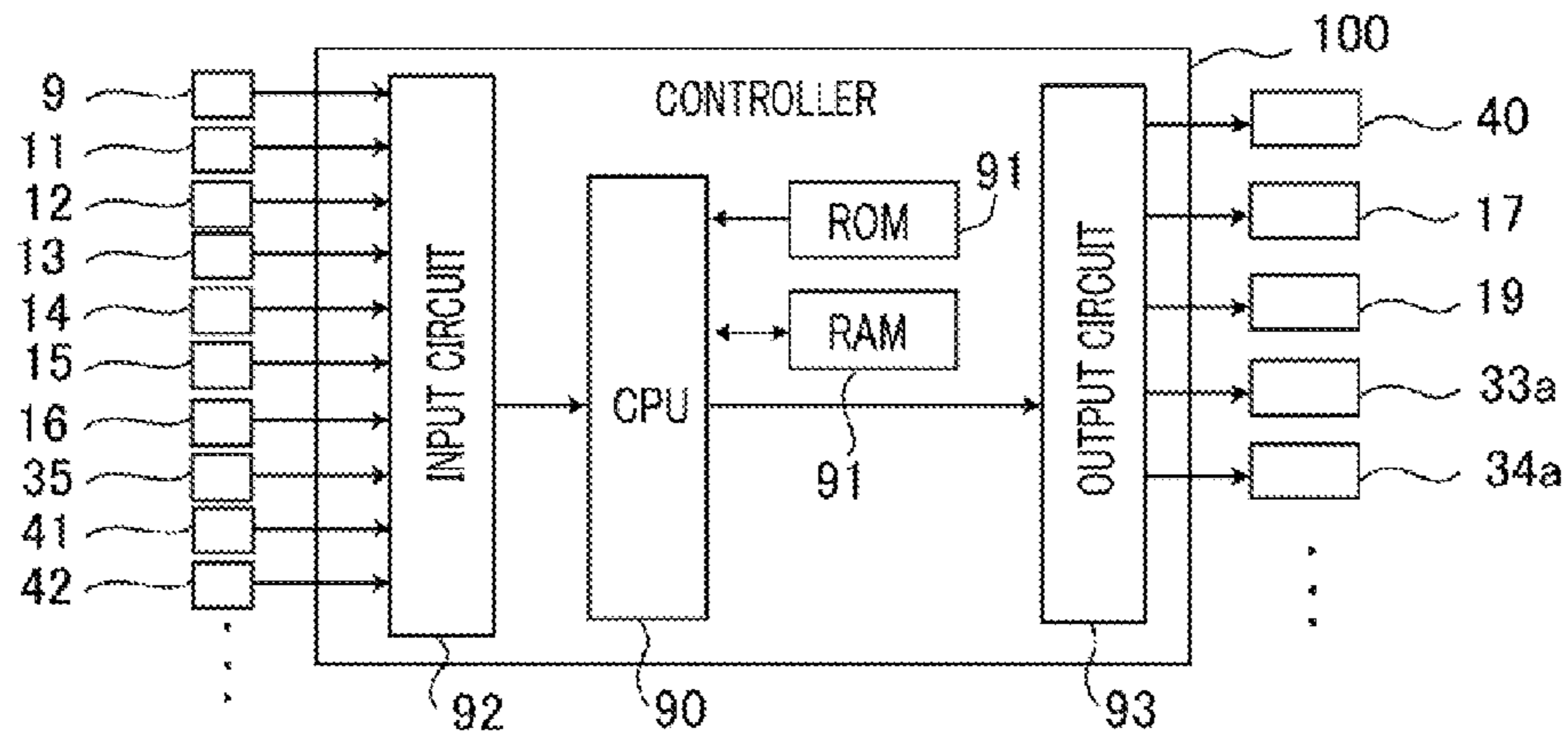


FIG. 4

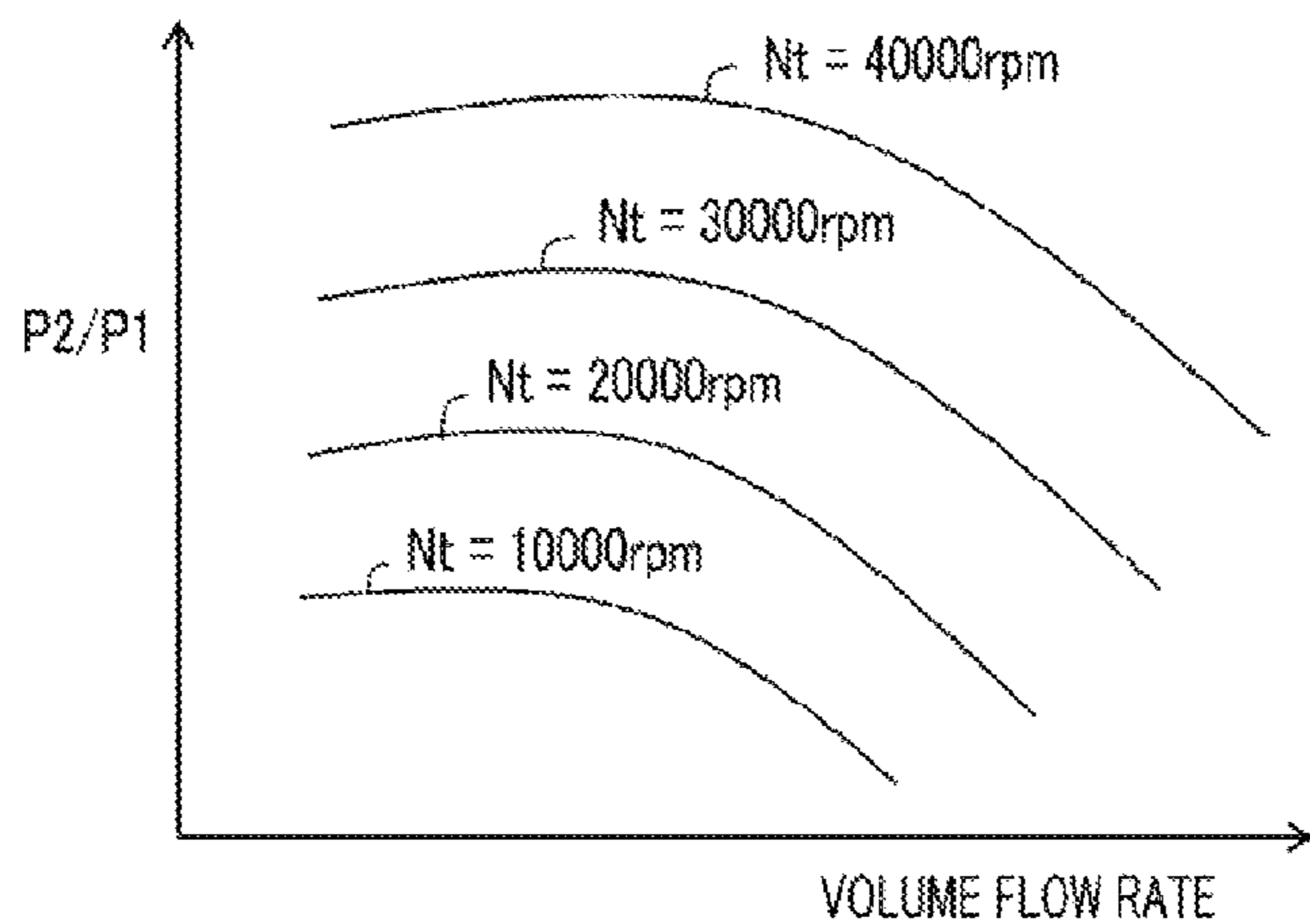


FIG. 5

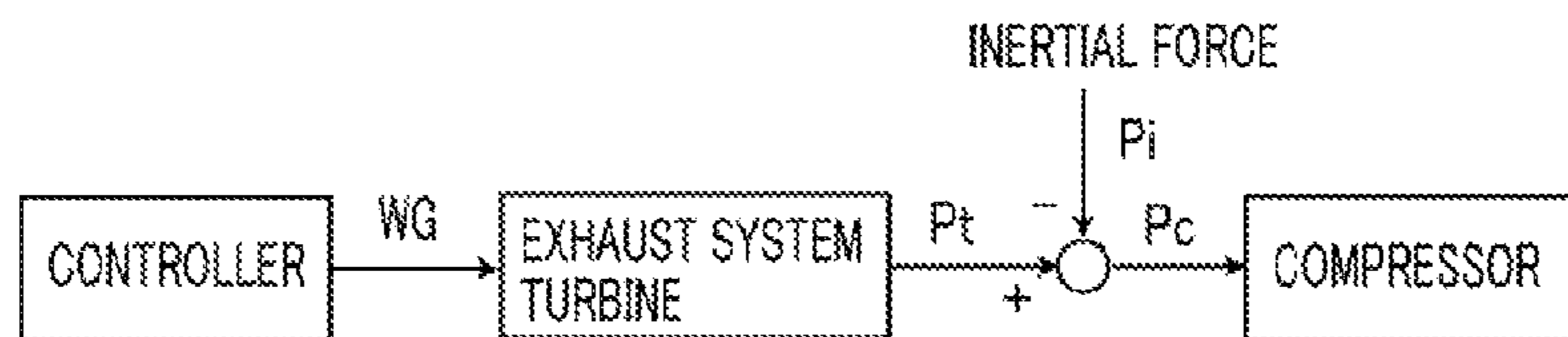


FIG. 6

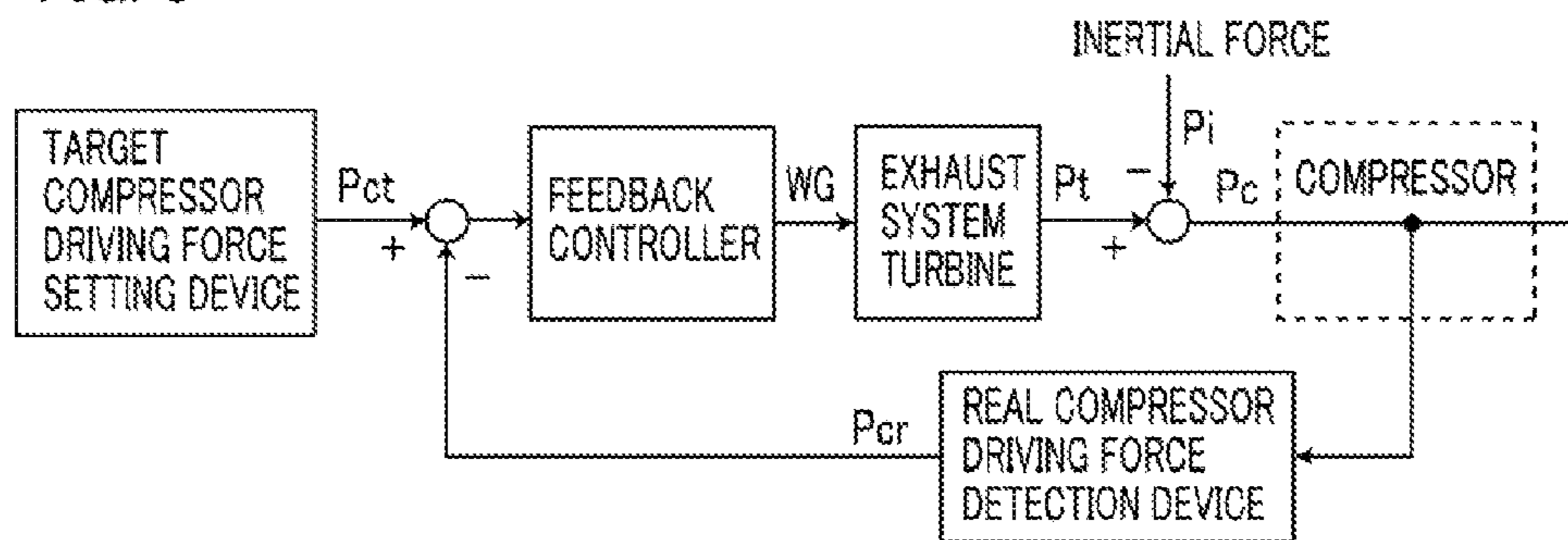


FIG. 7

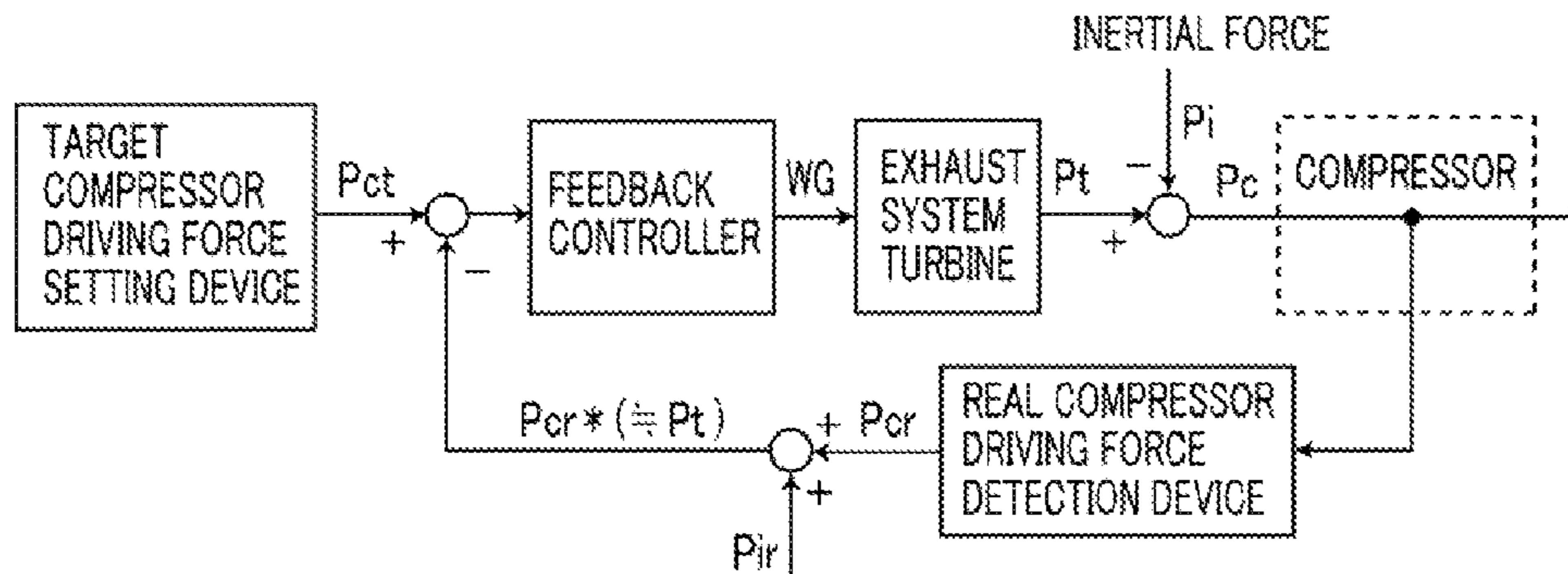


FIG. 8

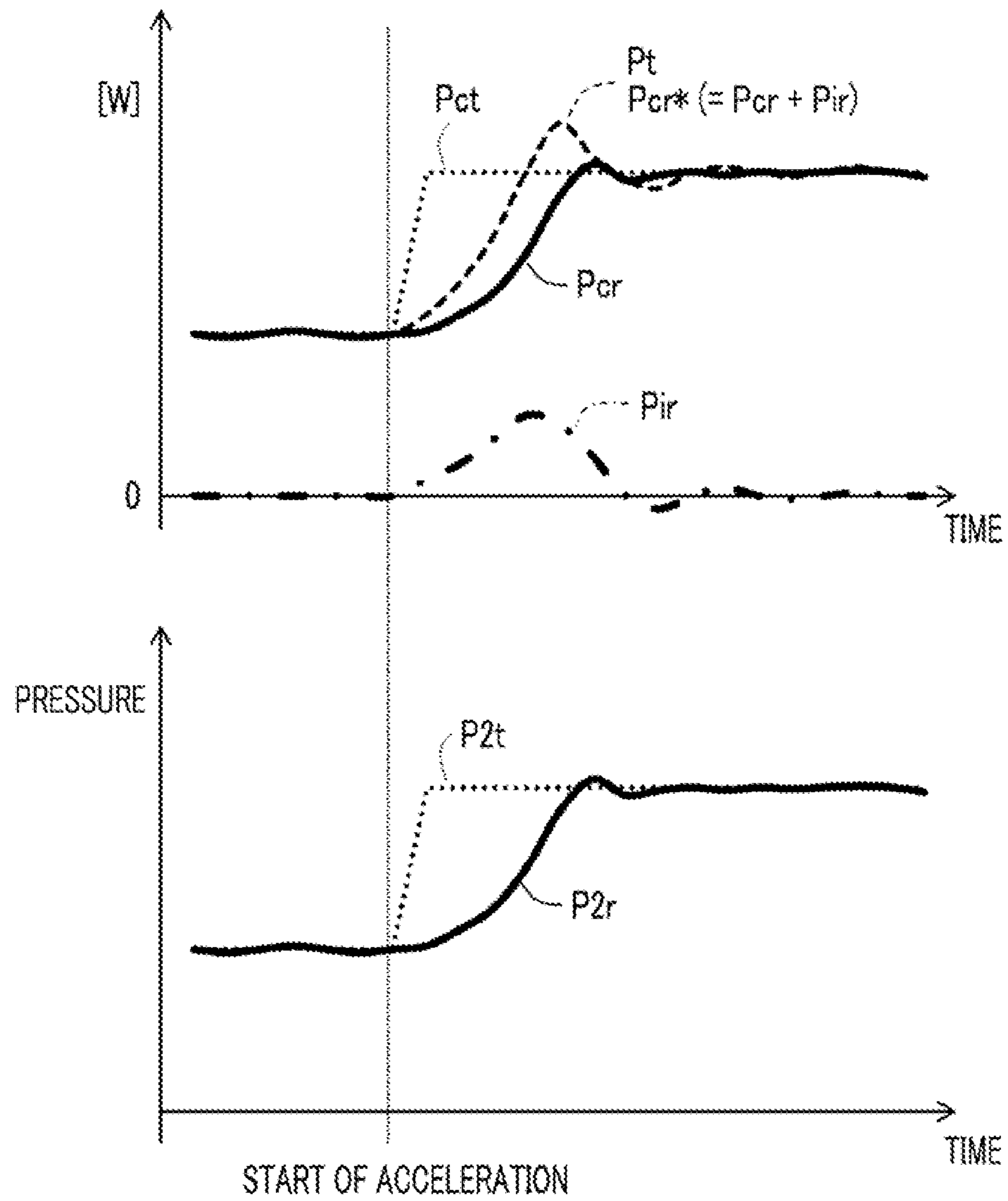


FIG. 9

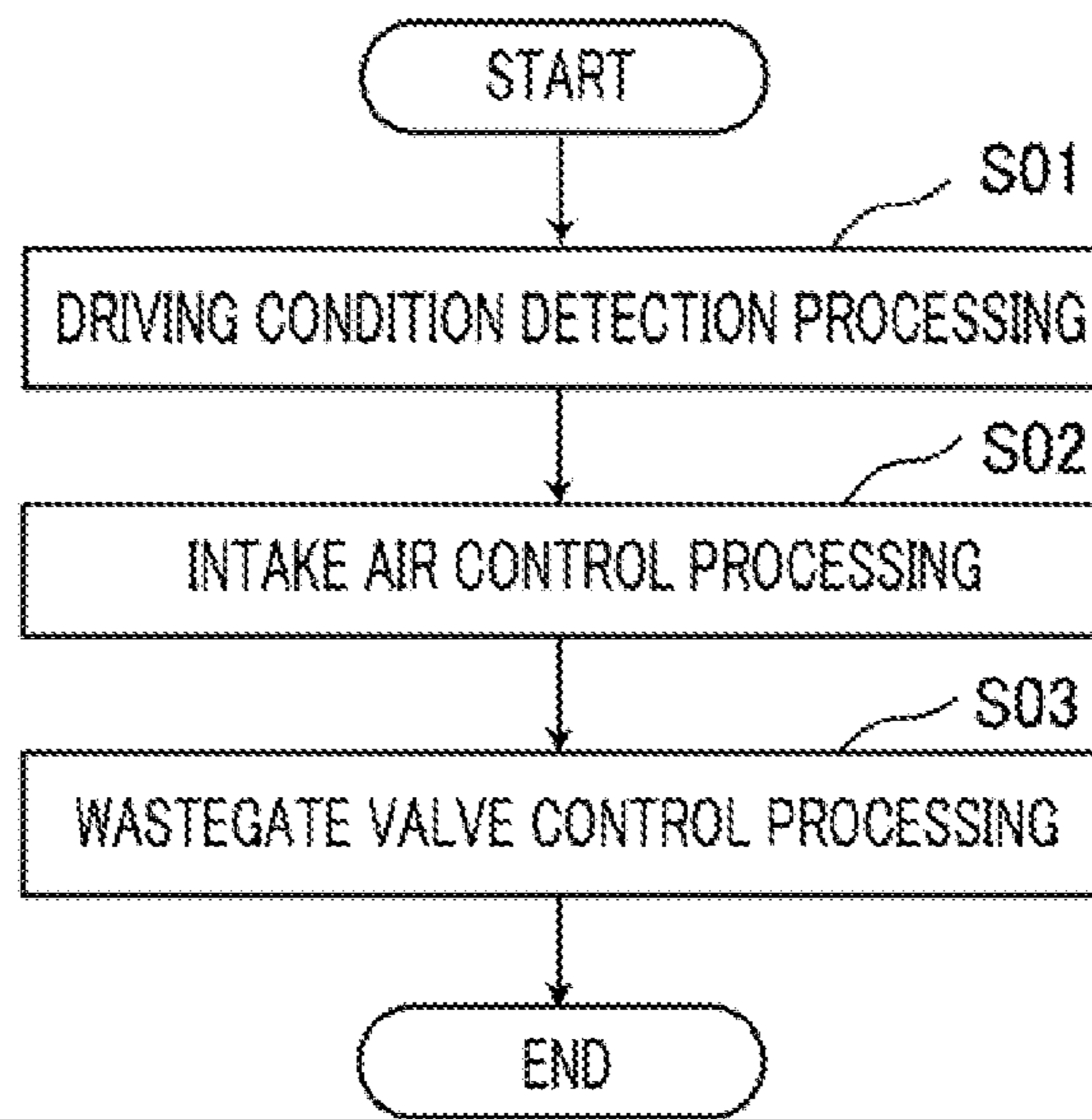


FIG. 10

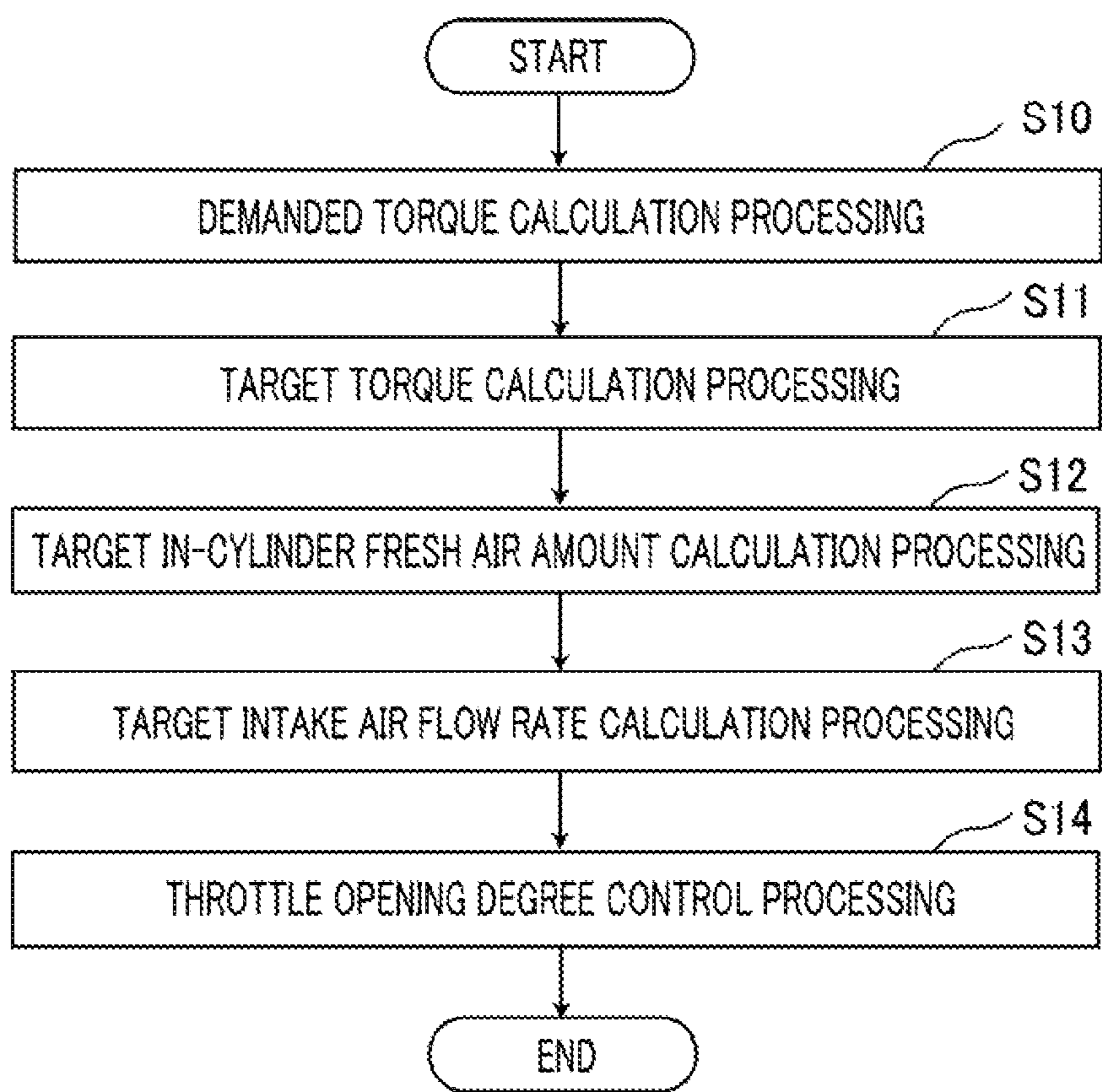


FIG. 11

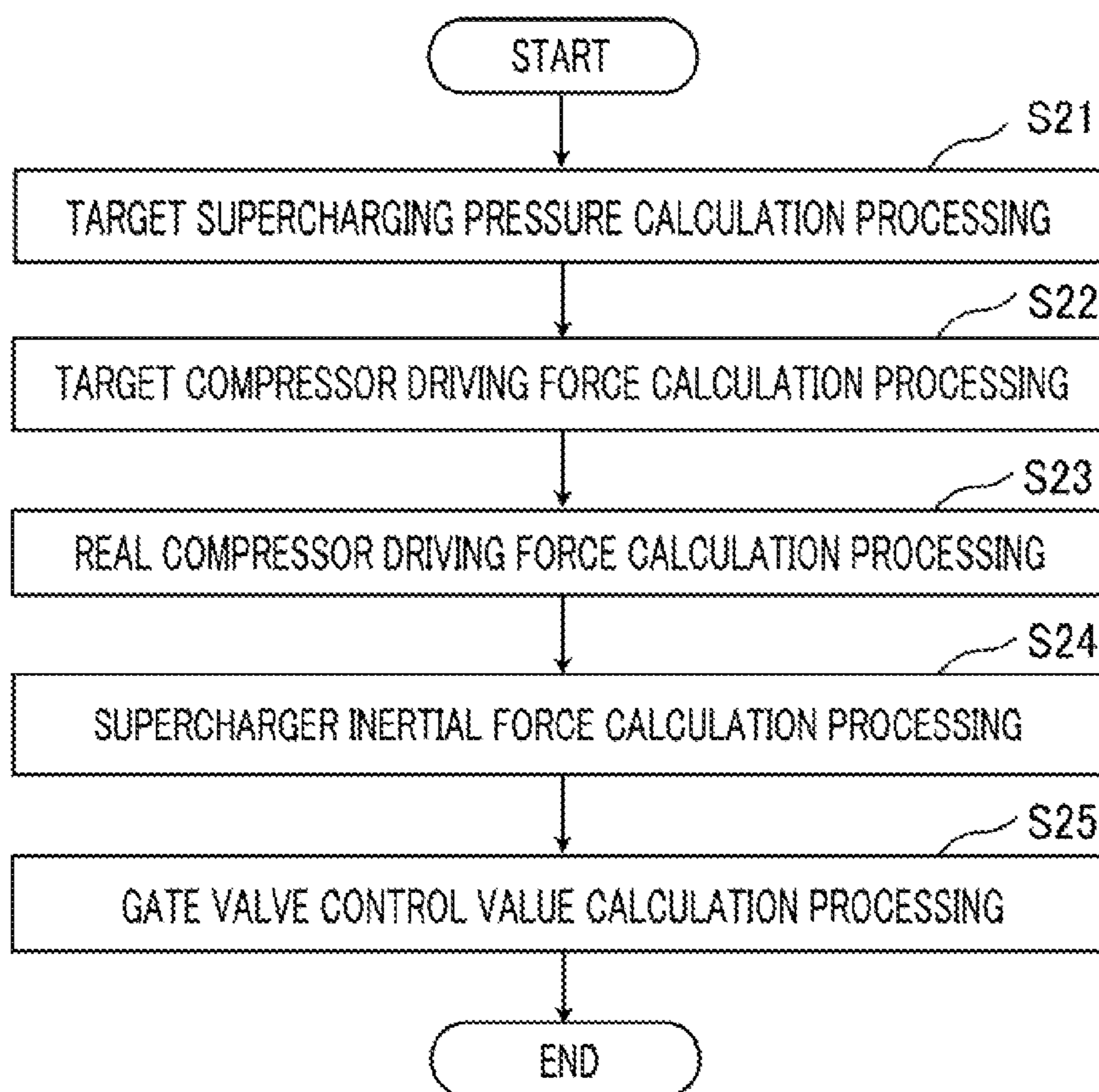


FIG. 12

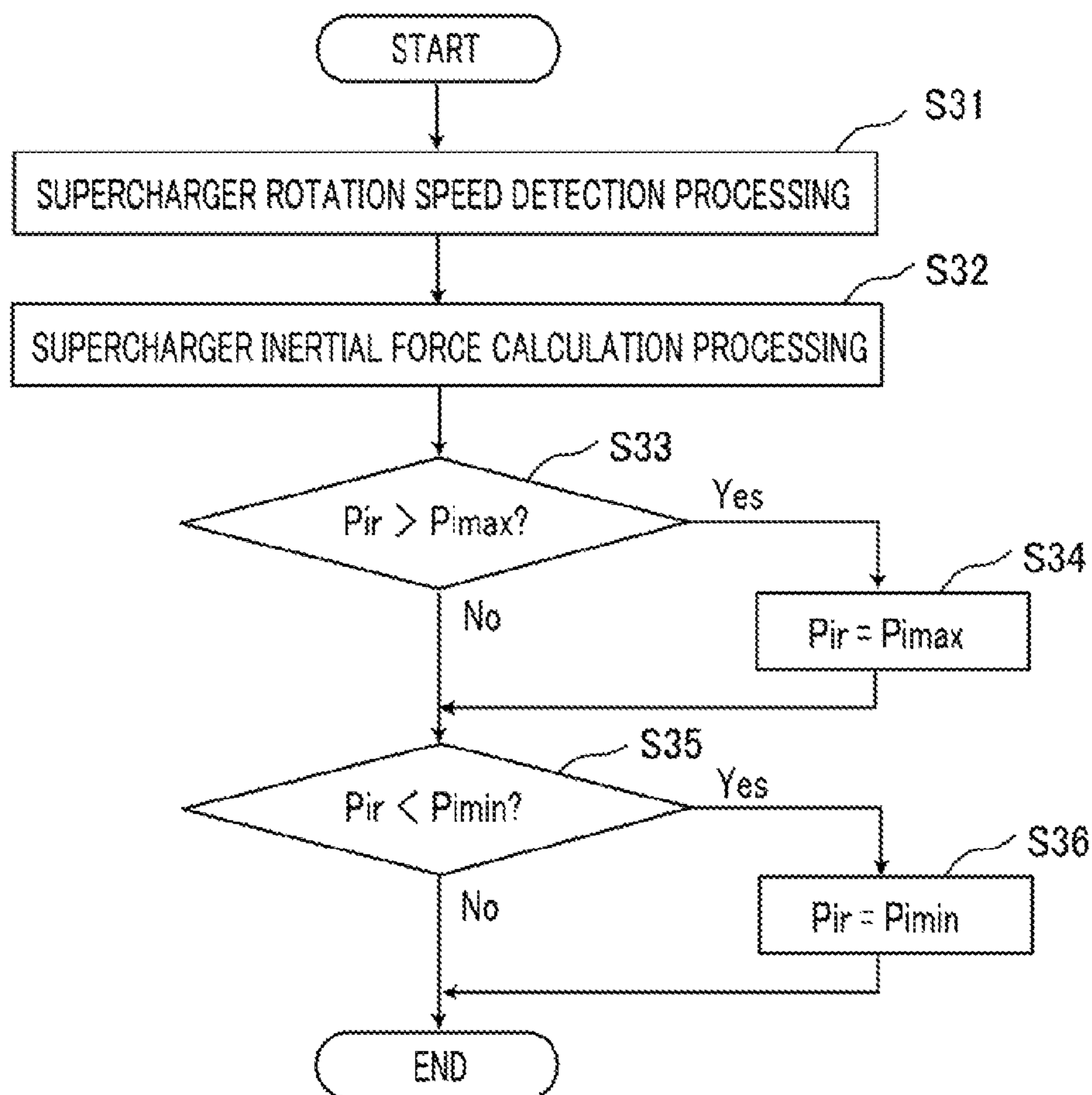


FIG. 13

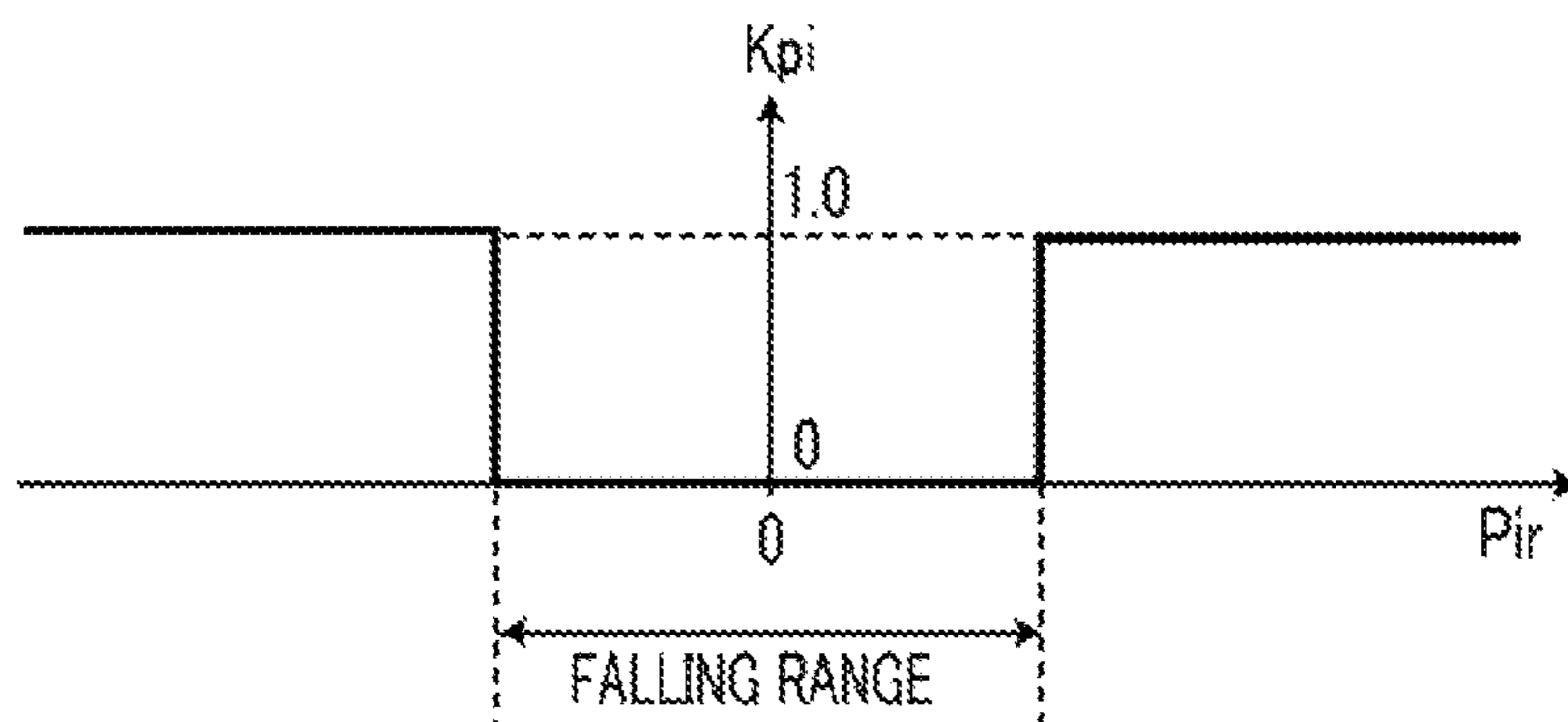
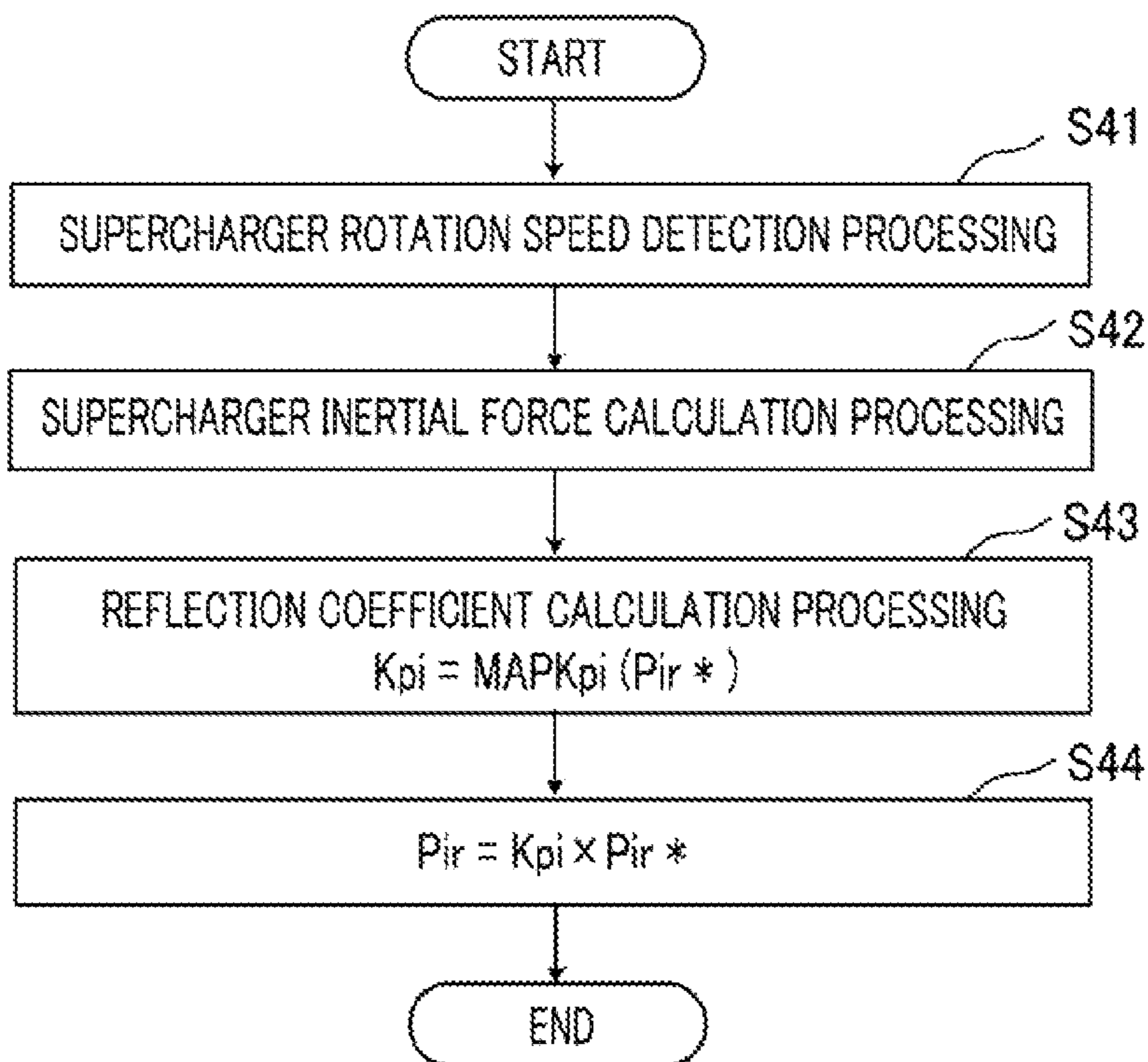


FIG. 14



1

**CONTROLLER FOR
SUPERCHARGER-EQUIPPED INTERNAL
COMBUSTION ENGINE**

INCORPORATION BY REFERENCE

The disclosure of Japanese Patent Application No. 2015-163410 filed on Aug. 21, 2015 including its specification, claims and drawings, is incorporated herein by reference in its entirety.

BACKGROUND OF THE INVENTION

The present invention relates to a controller for an internal combustion engine equipped with a supercharger including an actuator for driving a wastegate valve.

DESCRIPTION OF THE RELATED ART

To date, there has been known a supercharger in which a compressor thereof, which rotates and drives a turbine with exhaust gas, is mounted in an intake path of an internal combustion engine, for the purpose of increasing the output of the internal combustion engine.

Because its supercharging pressure increases more than necessary in the state of a high rotation speed and a high load and hence the internal combustion engine may be broken, the supercharger, in general, has an exhaust gas bypass that bypasses the turbine; the supercharging pressure is controlled at an appropriate level in such a way that a wastegate valve provided in the exhaust gas bypass is opened so that part of exhaust gas is made to enter the exhaust gas bypass in a ramifying manner and hence the amount of the exhaust gas flowing into the turbine is adjusted (e.g., refer to Japanese Patent Application No. JP-A-H09-228848).

As described above, the exhaust pressure and the supercharging pressure of the supercharger are controlled based on the opening degree of the wastegate valve. The control amount of the wastegate valve is determined through closed-loop control or simple open-loop control of an intake-system target amount (e.g., a target supercharging pressure or a target intake amount) to be set based on the rotation speed and the load of the internal combustion engine.

Meanwhile, in recent years, there has been proposed an internal combustion engine controller in which as the output target value of an internal combustion engine, the output axle torque of the internal combustion engine, which is a demanded value of driving force demanded by a driver or from the vehicle side, is utilized and then the air amount, the fuel amount, and the ignition timing, which are control amounts of the internal combustion engine, are determined so that an excellent traveling performance can be obtained. Moreover, it is known that among control amounts for an internal combustion engine, an air amount is a most influential control amount for the output axle torque of the internal combustion engine; thus, there has been provided an internal combustion engine controller that accurately controls air amount (e.g., refer to Japanese Patent Application No. JP-A-2009-013922).

Furthermore, there has been proposed a method in which the conventional wastegate valve control apparatus disclosed in JP-A-H09-228848 is applied to an internal combustion engine controller, such as the one disclosed in JP-A-2009-013922, that determines the output target value of an internal combustion engine. For example, in the technology disclosed in Japanese Patent No. JP-5420013, listed below, a target intake air flow rate (\approx a target charging

2

efficiency) is calculated based on the output target value of an internal combustion engine; a target supercharging pressure is calculated based on the target charging efficiency and the rotation speed; based on the target intake air flow rate and the target supercharging pressure, a target compressor driving force required for driving the supercharger is calculated; then, by use of the characteristic (FIG. 9 in JP-5420013) that the relationship between the exhaust gas flow rate and the compressor driving force (the turbine output) changes in accordance with the control value for the actuator of the wastegate valve, a control value for the actuator of the wastegate valve is calculated based on the exhaust gas flow rate and the target compressor driving force.

SUMMARY OF THE INVENTION

In the technology disclosed in JP-5420013, the control value for the actuator of the wastegate valve is calculated based on the exhaust gas flow rate and the target compressor driving force in a feedforward manner, and the control value for the actuator is changed in a feedback manner so that a real compressor driving force approaches the target compressor driving force. However, in the technology disclosed in JP-5420013, inertial force produced by the inertial moment of the supercharger is not taken into consideration. Accordingly, in the case where the inertial force of the supercharger becomes large in transient driving, no stable feedback response for the real compressor driving force can be obtained; thus, there has been a problem that the overshooting amount and the undershooting amount of the real compressor driving force becomes large.

The present invention has been implemented in order to solve the foregoing problem; the objective thereof is to provide a controller for an internal combustion engine equipped with a supercharger that can improve the feedback response of compressor driving force.

A controller for a supercharger-equipped internal combustion engine according to the present invention is a controller for an internal combustion engine equipped with a supercharger having a turbine provided in an exhaust path, a compressor that is provided at the upstream side of a throttle valve in an intake path and rotates integrally with the turbine, a wastegate valve provided in a bypass path, of the exhaust path, that bypasses the turbine, and a gate valve actuator that drives the wastegate valve; the control method for an internal combustion engine equipped with a supercharger includes

a driving-condition detector that detects a real rotation speed of the supercharger,

a target compressor driving force calculator that calculates a target compressor driving force, which is a target value of driving force for the compressor,

a real compressor driving force calculator that calculates real compressor driving force, which is a real value of driving force for the compressor,

a supercharger inertial force calculator that calculates inertial force produced by an inertial moment of the supercharger, based on the real rotation speed of the supercharger, and

a gate valve control value calculator that implements driving force feedback control for changing a gate valve control value, which is a control value for the gate valve actuator, so that an addition value obtained by adding the inertial force to the real compressor driving force

approaches the target compressor driving force, and performs driving control of the wastegate valve based on the gate valve control value.

The controller for a supercharger-equipped internal combustion engine according to the present invention makes it possible to implement feedback control of compressor driving force while taking inertial force produced by an inertial moment of the supercharger into consideration. Accordingly, even in the case where the inertial force of the supercharger becomes large in transient driving, a stable feedback response for the real compressor driving force can be obtained; thus, the overshooting amount and the undershooting amount of the real compressor driving force can be reduced.

The foregoing and other object, features, aspects, and advantages of the present invention will become more apparent from the following detailed description of the present invention when taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic configuration diagram of a supercharger-equipped internal combustion engine according to Embodiment 1 of the present invention;

FIG. 2 is a block diagram of a controller for the supercharger-equipped internal combustion engine according to Embodiment 1 of the present invention;

FIG. 3 is a hardware configuration diagram of the controller for the supercharger-equipped internal combustion engine according to Embodiment 1 of the present invention;

FIG. 4 is a graph for explaining a map utilized in estimation of a rotation speed of a supercharger according to Embodiment 1 of the present invention.

FIG. 5 is a block diagram for explaining the control system of a wastegate valve according to Embodiment 1 of the present invention;

FIG. 6 is a block diagram for explaining a feedback control system according to a comparative example of the present invention;

FIG. 7 is a block diagram for explaining a feedback control system according to Embodiment 1 of the present invention;

FIG. 8 is a timing chart for explaining a control behavior according to Embodiment 1 of the present invention;

FIG. 9 is a flowchart representing the processing by the controller according to Embodiment 1 of the present invention;

FIG. 10 is a flowchart representing the processing by an intake air control unit according to Embodiment 1 of the present invention;

FIG. 11 is a flowchart representing the processing by a wastegate valve control unit according to Embodiment 1 of the present invention;

FIG. 12 is a flowchart representing the processing by a supercharger inertial force calculation unit according to Embodiment 2 of the present invention;

FIG. 13 is a graph for explaining a reflection coefficient map according to Embodiment 3 of the present invention; and

FIG. 14 is a flowchart representing the processing by a supercharger inertial force calculation unit according to Embodiment 3 of the present invention.

DETAILED DESCRIPTION OF THE EMBODIMENTS

1. Embodiment 1

A controller **100** for an internal combustion engine **1** equipped with a supercharger **36** (hereinafter, referred to simply as the controller **100**) according to Embodiment 1 will be explained with reference to the drawings. FIG. 1 is a schematic configuration diagram of the internal combustion engine **1** (hereinafter, referred to as the engine **1**) equipped with the supercharger **36**; FIG. 2 is a block diagram of the controller **100** according to Embodiment 1. 1-1. The Configuration of the Engine 1

At first, the configuration of the engine **1** will be explained. As illustrated in FIG. 1, the engine **1** has a cylinder **8** in which a fuel-air mixture is combusted. The engine **1** and the controller **100** are mounted in a vehicle; the engine **1** functions as a driving force source for the vehicle (wheels). The engine **1** has an intake path **2** for supplying air to the cylinder **8** and an exhaust path **7** for discharging exhaust gas from the cylinder **8**. The intake path **2** is formed of an intake pipe and the like; the exhaust path **7** is formed of an exhaust pipe and the like. The intake path **2** has an intake manifold **5** for supplying air to the respective cylinders **8**. A throttle valve **4** is provided at a position, in the intake path **2**, that is at the upstream side of the intake manifold **5**. Accordingly, the intake path **2** at the downstream side of the throttle valve **4** is formed of the intake manifold **5**. The engine **1** has the supercharger **36**. The supercharger **36** includes a turbine **32** provided in the exhaust path **7**, a compressor **31** that is provided at a position, in the intake path **2**, that is at the upstream side of the throttle valve **4**, and rotates integrally with the turbine **32**, a bypass **37** (hereinafter, referred to as an exhaust gas bypass **37**), of the exhaust path **7**, that bypasses the turbine **32**, a wastegate valve **34** provided in the exhaust gas bypass **37**, and a gate valve actuator **34a** that drives the wastegate valve **34**. The exhaust gas bypass **37** is a bypass flow path, for the turbine **32**, that connects the portion, of the exhaust path **7**, that is at the upstream side of the turbine **32** with the portion, of the exhaust path **7**, that is at the downstream side of the turbine **32**. The wastegate valve **34** is a valve for changing the flow path area (opening degree) of the exhaust gas bypass **37**.

When exhaust gas drives and rotates the turbine **32**, the compressor **31** rotates integrally with the turbine **32**, compresses air in the intake path **2**, and transmits the air to the cylinder **8**. The turbine **32** and the compressor **31** are coupled with each other by a turbine shaft **39** in such a way as to integrally rotate on the same axis. A rotation speed sensor **42** that generates an electric signal corresponding to the rotation speed of the turbine shaft **39** is provided on the turbine shaft **39**. When the opening degree of the wastegate valve **34** is increased through the gate valve actuator **34a**, a wastegate flow rate, out of the exhaust gas flow rate Q_{ex} to be exhausted from the engine **1** (cylinder **8**), increases; the wastegate flow rate is the flow rate of part of the exhaust gas, which bypasses the turbine **32** so as to flow in the exhaust gas bypass **37**. As a result, a turbine flow rate, which is the flow rate of exhaust gas that flows into the turbine **32**, decreases. Accordingly, the rotation driving forces of the turbine **32** and the compressor **31** are weakened. The gate valve actuator **34a** is an electric one that changes the opening degree of the wastegate valve **34** through the rotation driving force of an electric motor. The gate valve actuator **34a** may be a pressure-type one in which a diaphragm is supplied with a pressure obtained by reducing a supercharging pressure P_2 by a decompression amount

adjusted through a solenoid valve and then the driving force of the diaphragm changes the opening degree of the wastegate valve **34**.

In Embodiment 1, the supercharger **36** includes a bypass **38** (hereinafter, referred to as an air bypass **38**), of the intake path **2**, that bypasses the compressor **31**, an air bypass valve **33** provided in the air bypass **38**, and a bypass valve actuator **33a** that drives the air bypass valve **33**. The bypass valve actuator **33a** is a pressure-type one having a diaphragm that is actuated by the pressure difference between the supercharging pressure **P2** and a manifold pressure **Pb**. When the supercharging pressure **P2** increases to exceed a predetermined pressure difference from the manifold pressure **Pb**, the diaphragm is activated and hence the air bypass valve **33** is opened; thus, the upstream side and the downstream side of the compressor **31** are connected. Accordingly, it is made possible to prevent mechanical damage to an intake pipe or the like caused by an abnormal rise of the supercharging pressure **P2** at a time when the accelerator pedal is released. While an after-mentioned wastegate valve control unit **112** controls the opening degree of the wastegate valve **34**, the air bypass valve **33** is basically closed.

An air cleaner **3** for purifying taken-in outer air is mounted at the most upstream side of the intake path **2**. At a position that is at the downstream side (the side closer to the cylinder **8**) of the air cleaner **3** in the intake path **2** and at the upstream side of the compressor **31**, an air flow sensor **12**, which generates an electric signal corresponding to an intake air flow rate **Qa**, and an intake-air temperature sensor **13**, which generates an electric signal corresponding to an intake-air temperature **T1** in the intake path **2**, are provided, as discrete components or as an integrated component (as an integrated component, in this example). An atmospheric pressure sensor **9**, which generates an electric signal corresponding to an atmospheric pressure **P1**, is provided at a position that is at the downstream side of the air cleaner **3** in the intake path **2** and at the upstream side of the compressor **31**. The pressure at the upstream side of the compressor **31** can be regarded as being equal to the atmospheric pressure **P1**. The atmospheric pressure sensor **9** may be contained in the controller **100**.

An exhaust gas purification catalyst **22** is provided at the downstream side of the turbine **32** in the exhaust path **7**. At a position that is at the downstream side of the turbine **32** in the exhaust path **7** and the upstream side (the side closer to the cylinder **8**) of the exhaust gas purification catalyst **22**, there is provided an air-fuel ratio sensor **16** that generates an electric signal corresponding to an air-fuel ratio **AF**, which is the ratio of air to fuel in a combustion gas.

An intercooler **30** for cooling compressed air is provided at the downstream side of the compressor **31** in the intake path **2**. The throttle valve **4** for adjusting an air amount to be taken in by the engine **1** is provided at the downstream side of the intercooler **30**. The throttle valve **4** is opened or closed by a throttle motor (a motor for driving the throttle valve) **40**. The throttle valve **4** is connected with a throttle position sensor **14** that generates an electric signal corresponding to a throttle opening degree, which is the opening degree of the throttle valve **4**. In a supercharging intake path, which is part, of the intake path **2**, that is at downstream side of the compressor **31** and at the upstream side of the throttle valve **4**, there is provided a supercharging pressure sensor **35** that generates an electric signal corresponding to the supercharging pressure **P2**, which is the pressure of air in the supercharging intake path.

The part, of the intake path **2**, that is at the downstream side of the throttle valve **4** constitutes the intake manifold **5**,

which functions also as a surge tank for suppressing an intake-air ripple. A manifold pressure sensor **15** that generates an electric signal corresponding to the manifold pressure **Pb**, which is the pressure of air in the intake manifold **5**, is provided in the intake manifold **5**. Unlike Embodiment 1 in which both the air flow sensor **12** and the manifold pressure sensor **15** are provided, the supercharger-equipped internal combustion engine may be provided only with the manifold pressure sensor **15** but with no air flow sensor **12**. In the case where only the manifold pressure sensor **15** is provided, it may be allowed that the intake-air temperature sensor **13** is provided in the intake manifold **5** so as to detect the intake-air temperature inside the intake manifold **5**.

An injector **17** for injecting a fuel is provided at the downstream side (the side closer to the cylinder **8**) of the intake manifold **5**. The injector **17** may be provided in such a way as to inject a fuel directly into the cylinder **8**.

In the top portion of the cylinder **8**, there are provided an ignition plug **18** for igniting an inflammable fuel-air mixture produced by mixing air taken into the cylinder **8** with a fuel injected from the injector **17** and an ignition coil **19** for generating energy with which the ignition plug **18** throws sparks. There are also provided an intake valve **20** for adjusting the intake air amount to be taken from the intake path **2** into the cylinder **8** and an exhaust valve **21** for adjusting the exhaust gas amount to be exhausted from the cylinder **8** to the exhaust path **7**. On the crankshaft of the engine **1**, there is provided a crank angle sensor **11** for generating an electric signal corresponding to the rotation angle of the engine **1**.

1-2. The Configuration of the Controller **100**

Next, the configuration of the controller **100** will be explained. The controller **100** is a controller whose control subject is the engine **1** equipped with the supercharger **36**. Respective control units **110** through **112** and the like provided in the controller **100** are realized by processing circuits included in the controller **100**. Specifically, as illustrated in FIG. **3**, the controller **100** includes, as processing circuits, a computing processing unit (computer) **90** such as a CPU (Central Processing Unit), storage apparatuses **91** that exchange data with the computing processing unit **90**, an input circuit **92** that inputs external signals to the computing processing unit **90**, an output circuit **93** that outputs signals from the computing processing unit **90** to the outside, and the like. As the storage apparatuses **91**, there are provided a RAM (Random Access Memory) that can read data from and write data in the computing processing unit **90**, a ROM (Read Only Memory) that can read data from the computing processing unit **90**, and the like. The input circuit **92** is connected with various kinds of sensors and switches and is provided with an A/D converter, an input port, and the like for inputting output signals from the sensors and the switches to the computing processing unit **90**. The output circuit **93** is connected with electric loads and is provided with a driving circuit, an output port, and the like for outputting a control signal from the computing processing unit **90** to the electric loads. In addition, the computing processing unit **90** runs software items (programs) stored in the storage apparatus **91** such as a ROM and collaborates with other hardware devices in the controller **100**, such as the storage apparatus **91**, the input circuit **92**, and the output circuit **93**, so that the respective functions of the control units **110** through **112** included in the controller **100** are realized. Setting data items such as maps and setting values to be utilized in the control units **110** through **112** are stored, as part of software items (programs), in the storage apparatus **91** such as a ROM.

In Embodiment 1, the input circuit **92** is connected with various kinds of sensors such as the atmospheric pressure sensor **9**, the crank angle sensor **11**, the air flow sensor **12**, the intake-air temperature sensor **13**, the throttle position sensor **14**, the manifold pressure sensor **15**, the air-fuel ratio sensor **16**, the supercharging pressure sensor **35**, an accelerator position sensor **41** for generating an electric signal corresponding to an accelerator operating amount, and rotation speed sensor **42** in the supercharger **36**. The output circuit **93** is connected with various kinds of actuators such as the throttle motor **40**, the injector **17**, the ignition coil **19**, the bypass valve actuator **33a**, and the gate valve actuator **34a**. Although not illustrated, the input circuit **92** is connected with a sensor for controlling the combustion in the engine **1** and sensors (e.g., a vehicle speed sensor, a water temperature sensor, and the like) for controlling the behavior of the vehicle.

As basic control, the controller **100** calculates the fuel injection amount and the ignition timing, based on inputted output signals and the like from the various kinds of sensors so as to perform driving control of a fuel ignition apparatus, an ignition apparatus, and the like (unillustrated). Although the details will be explained later, based on the output signal of the accelerator position sensor **41** and the like, the controller **100** calculates a demanded output torque demanded on the engine **1**, and then controls the throttle valve **4**, the wastegate valve **34**, and the like so that an intake air amount for realizing the demanded output torque is obtained.

1-2-1. Driving-Condition Detection Unit **110**

The controller **100** is provided with a driving-condition detection unit **110** (corresponding to a driving-condition detector) that detects the driving conditions of the engine **1** and the vehicle. The driving-condition detection unit **110** detects a real rotation speed N_{tr} of the supercharger **36** (the compressor **31** and the turbine **32**). In Embodiment 1, the driving-condition detection unit **110** detects the real rotation speed N_{tr} of the supercharger **36**, based on the output signal of the rotation speed sensor **42** provided in the supercharger **36**.

Alternatively, the driving-condition detection unit **110** may detect the real rotation speed N_{tr} of the supercharger **36**, based on a real intake air flow rate Q_{ar} and a real before/after-compressor pressure ratio P_{2r}/P_{1r} , which is the pressure ratio of a real supercharging pressure P_{2r} and a real atmospheric pressure P_{1r} . Specifically, by use of a rotation speed map, as represented in FIG. 4, in which the relationship among the intake-air volume flow rate obtained by dividing the intake air flow rate Q_a by the air density, a before/after-compressor pressure ratio P_2/P_1 , which is the pressure ratio of the supercharging pressure P_2 and the atmospheric pressure P_1 , and a rotation speed N_t of the supercharger **36** is preliminarily set, the driving-condition detection unit **110** calculates the real rotation speed N_{tr} of the supercharger **36** corresponding to a real volume flow rate obtained by dividing the real intake air flow rate Q_{ar} by the air density and the real before/after-compressor pressure ratio P_{2r}/P_{1r} . FIG. 4 represents an iso-rotation speed line obtained by connecting points at which the respective rotation speeds N_t become equal to one another when the volume flow rate and the before/after-compressor pressure ratio P_2/P_1 are changed.

The driving-condition detection unit **110** detects the real rotation speed N_{er} of the engine **1**, the real intake air flow rate Q_{ar} , and the real atmospheric pressure P_{1r} . Specifically, the driving-condition detection unit **110** detects the real rotation speed N_{er} of the engine **1**, based on the output signal

of the crank angle sensor **11**, detects the real intake air flow rate Q_{ar} of the engine **1**, based on the output signal of the air flow sensor **12** or the manifold pressure sensor **15**, and detects the real atmospheric pressure P_{1r} , based on the output signal of the atmospheric pressure sensor **9**.

In addition to the foregoing driving conditions, the driving-condition detection unit **110** detects various kinds of driving conditions such as a real intake air temperature T_{1r} , a real throttle opening degree TH_r , a real manifold pressure P_{br} , an exhaust gas air-fuel ratio AF , a real supercharging pressure P_{2r} , and an accelerator opening degree D . Specifically, the driving-condition detection unit **110** detects the real intake air temperature T_{1r} , based on the output signal of the intake-air temperature sensor **13**, detects the real throttle opening degree TH_r , based on the output signal of the throttle position sensor **14**, detects the real manifold pressure P_{br} , based on the output signal of the manifold pressure sensor **15**, detects the exhaust gas air-fuel ratio AF , based on the output signal of the air-fuel ratio sensor **16**, detects the real supercharging pressure P_{2r} , based on the output signal of the supercharging pressure sensor **35**, and detects the accelerator opening degree D , based on the output signal of the accelerator position sensor **41**.

<Real Intake Air Flow Rate Calculation Unit **141**>

The driving-condition detection unit **110** is provided with a real intake air flow rate calculation unit **141**. The real intake air flow rate calculation unit **141** calculates the real intake air flow rate Q_{ar} , which is the flow rate of air that is taken in by the engine **1** (the intake path **2**). In Embodiment 1, based on the output signal of the air flow sensor **12** or the manifold pressure sensor **15** (in this example, the air flow sensor **12**), the real intake air flow rate calculation unit **141** calculates the real intake air flow rate Q_{ar} [g/s].

<Real In-Cylinder Fresh Air Amount Calculation Unit **142**>

The driving-condition detection unit **110** is provided with a real in-cylinder fresh air amount calculation unit **142**. Based on the output signal of the air flow sensor **12** or the manifold pressure sensor **15** (in this example, the air flow sensor **12**), the real in-cylinder fresh air amount calculation unit **142** calculates a real charging efficiency E_{cr} and a real in-cylinder fresh air amount Q_{cr} .

In Embodiment 1, as represented in the equation (1) below, the real in-cylinder fresh air amount calculation unit **142** applies first-order-lag filter processing, which simulates a delay in the intake manifold **5** (surge tank), to the value obtained by multiplying the real intake air flow rate Q_{ar} by the stroke period ΔT (in this example, the interval of BTDC5degCA), in order to calculate the real in-cylinder fresh air amount Q_{cr} per stroke [g/stroke].

$$Q_{cr}(n) = KCCA \times Q_{cr}(n-1) + (1 - KCCA) \times Q_{ar}(n) \times \Delta T(n) \quad (1)$$

where $KCCA$ is a filter coefficient.

Alternatively, it may be allowed that as represented in the equation (2) below, the real in-cylinder fresh air amount calculation unit **142** calculates the volume of air, in the intake manifold **5**, that has been taken in by the cylinder **8**, by multiplying a volumetric efficiency K_v on the basis of the intake manifold **5** by a cylinder volume V_c , and then multiplies the calculated air volume by an air density ρ_b , which is calculated based on the real manifold pressure P_{br} and the real intake air temperature T_{1r} , in order to calculate the real in-cylinder fresh air amount Q_{cr} [g/stroke]. In the equation (2), the volumetric efficiency K_v is the ratio of the volume of air, in the intake manifold **5**, that is taken in by the cylinder, to the cylinder volume V_c ($K_v = \text{the volume of air in the intake manifold } 5 \text{ taken in by the cylinder } 8 / V_c$). By use of a map in which the relationship among the rotation

speed N_e , the manifold pressure P_b , and the volumetric efficiency K_v is preliminarily set, the real in-cylinder fresh air amount calculation unit **142** calculates the volumetric efficiency K_v corresponding to the real rotation speed N_e and the real manifold pressure P_b .

$$Q_{cr} = (K_v \times V_c) \times \rho_b, \rho_b = P_b / (R \times T_b) \quad (2)$$

where R is a gas constant.

As represented in the equation (3) below, the real in-cylinder fresh air amount calculation unit **142** calculates the real charging efficiency E_{cr} by dividing the real in-cylinder fresh air amount Q_{cr} by a value obtained by multiplying the density of air under the standard atmospheric condition by the cylinder volume V_c . The real charging efficiency E_{cr} is the ratio of the real in-cylinder fresh air amount Q_{cr} to the density ($\times V_c$) of air under the standard atmospheric condition, with which the cylinder volume V_c is filled. The standard atmospheric condition denotes the state of 1 atm and 25° C.

$$E_{cr} = Q_{cr} / (\rho_0 \times V_c) \quad (3)$$

1-2-2. Intake Air Control Unit **111**

The controller **100** is provided with an intake air control unit **111** (corresponding to an intake air controller) that controls intake air of the engine **1**. The intake air control unit **111** calculates a target intake air flow rate Q_{at} , which is a target value of the intake air flow rate Q_a , and a target charging efficiency E_{ct} , which is a target value of the charging efficiency E_c .

In Embodiment 1, the intake air control unit **111** includes a demanded torque calculation unit **120** that calculates a demanded output torque TRQ_d , which is the output torque demanded on the engine **1**, a target torque calculation unit **121** that calculates a target output torque TRQ_t or a target indicated mean effective pressure P_{it} , based on the demanded output torque TRQ_d , a target in-cylinder fresh air amount calculation unit **122** that calculates the target charging efficiency E_{ct} and the target in-cylinder fresh air amount Q_{ct} , based on the target output torque TRQ_t or the target indicated mean effective pressure P_{it} , a target intake air flow rate calculation unit **123** that calculates the target intake air flow rate Q_{at} , based on the target in-cylinder fresh air amount Q_{ct} , and a throttle opening degree control unit **124** that controls the throttle opening degree, based on the target intake air flow rate Q_{at} .

Hereinafter, the control units **120** through **124** in the intake air control unit **111** will be explained in detail.

<Demanded Torque Calculation Unit **120**>

The demanded torque calculation unit **120** calculates the demanded output torque TRQ_d , based on the accelerator opening degree D and a demand from an external controller. Based on the real rotation speed N_e of the engine **1** (or a traveling speed VS of the vehicle) and the accelerator opening degree D , the demanded torque calculation unit **120** calculates a driver-demanded output torque, which is an output torque, of the engine **1**, that is demanded by the driver of the vehicle. Specifically, by use of a map in which the relationship among the real rotation speed N_e of the engine **1** (or the traveling speed VS), the accelerator opening degree D , and the driver-demanded output torque is preliminarily set, the demanded torque calculation unit **120** calculates driver-demanded output torque corresponding to the real rotation speed N_e (or the traveling speed VS) and the accelerator opening degree D .

An external controller (e.g., a transmission controller, a brake controller, a controller for traction control, or the like) inputs an external demanded output torque TRR to the

controller **100**. In accordance with the driving condition, the demanded torque calculation unit **120** selects one of the driver-demanded output torque and the external demanded output torque TRR and then outputs the selected torque, as the demanded output torque TRQ_d . The demanded output torque TRQ_d denotes the demanded value of torque outputted from the crankshaft of the engine **1**.

<Target Torque Calculation Unit **121**>

The target torque calculation unit **121** calculates the target output torque TRQ_t or the target indicated mean effective pressure P_{it} , based on the demanded output torque TRQ_d . The target torque calculation unit **121** calculates a load of an engine auxiliary apparatus corresponding to the real driving condition such as the real rotation speed N_e , by use of a map in which the relationship between the driving condition such as the rotation speed N_e and the load of the engine auxiliary apparatus is preliminarily set, based on experimental data obtained by measuring the respective loads of various kinds of engine auxiliary apparatuses (e.g., an alternator, an air conditioner compressor, a power steering pump, a transmission pump, a torque converter, and the like). The target torque calculation unit **121** adds the load (an absolute value) of an engine auxiliary apparatus to the demanded output torque TRQ_d so as to output an engine demanded output torque at a time when the load of an engine auxiliary apparatus is taken into consideration.

Next, the target torque calculation unit **121** calculates engine loss corresponding to the real driving condition such as the real rotation speed N_e , by use of a map in which the relationship between the driving condition such as the rotation speed N_e and the engine loss is preliminarily set, based on real data obtained by measuring mechanical loss and pumping loss inherent in the engine **1** (collectively, referred to as engine loss). Then, the target torque calculation unit **121** adds the engine loss (an absolute value) to the engine demanded output torque so as to calculate the target indicated mean effective pressure P_{it} to be produced in the cylinder **8**. It may be allowed that the target torque calculation unit **121** calculates the target output torque TRQ_t , instead of the target indicated mean effective pressure P_{it} .

<Target In-Cylinder Fresh Air Amount Calculation Unit **122**>

The target in-cylinder fresh air amount calculation unit **122** calculates the target in-cylinder fresh air amount Q_{ct} and the target charging efficiency E_{ct} , based on the target indicated mean effective pressure P_{it} or the target output torque TRQ_t . The target in-cylinder fresh air amount calculation unit **122** calculates the target in-cylinder fresh air amount Q_{ct} [g/stroke] and the target charging efficiency E_{ct} , based on the target indicated mean effective pressure P_{it} or the target output torque TRQ_t , the target value of the air-fuel ratio AF , and the thermal efficiency η . By use of a map in which the relationship among the rotation speed N_e , the charging efficiency E_c , and the thermal efficiency η is preliminarily set, the target in-cylinder fresh air amount calculation unit **122** calculates the thermal efficiency η corresponding to the real rotation speed N_e and the real charging efficiency E_{cr} . The cylinder volume V_c denotes a stroke volume [L] per one cylinder of the cylinder **8**.

As represented in the equation (4) below, the target in-cylinder fresh air amount calculation unit **122** calculates the target in-cylinder fresh air amount Q_{ct} and the target charging efficiency E_{ct} , based on the target indicated mean

11

effective pressure P_{it} , the target value of the air-fuel ratio AF , and the thermal efficiency η .

$$Q_{ct} = AF \times P_{it} \times V_c / (\eta \times 44000)$$

$$E_{ct} = AF \times P_{it} / (\eta \times 44000 \times) \quad (4)$$

where “44000” is a heat generation amount per unit mass [J/kg] of a fuel (in this example, gasoline) to be utilized in the engine **1**. By use of a map in which the relationship among the rotation speed N_e , the charging efficiency E_c , and the thermal efficiency η is preliminarily set, the target in-cylinder fresh air amount calculation unit **122** calculates the thermal efficiency η corresponding to the real rotation speed N_{er} and the real charging efficiency E_{cr} .

It may be allowed that the target in-cylinder fresh air amount calculation unit **122** calculates the target charging efficiency E_{ct} by dividing the target in-cylinder fresh air amount Q_{ct} by the preliminarily set mass ($\rho_0 \times V_c$) of air with which the cylinder volume V_c is filled under the standard atmospheric condition. The target charging efficiency E_{ct} and the target in-cylinder fresh air amount Q_{ct} are values that correlate to each other; based on the calculated value of one of them, the value of the other one is calculated. <Target Intake Air Flow Rate Calculation Unit **123**>

Based on the target in-cylinder fresh air amount Q_{ct} , the target intake air flow rate calculation unit **123** calculates the target intake air flow rate (amount) Q_{at} [g/s] to be taken in by the engine **1** through the intake path **2**. In Embodiment 1, as represented in the equation (5) below, the target intake air flow rate calculation unit **123** obtains a value by applying first-order advance filtering processing, which has a characteristic reverse to that of the foregoing first-order lag filtering processing represented in the equation (1), to the target in-cylinder fresh air amount Q_{ct} ; then, the target intake air flow rate calculation unit **123** divides the obtained value by the stroke period ΔT so as to calculate the target intake air flow rate Q_{at} . The target intake air flow rate Q_{at} corresponds to the target value of the flow rate of air that passes through the intake path **2** (for example, the throttle valve **4**) at the upstream side of the intake manifold **5** (the surge tank). In this example, the stroke period ΔT is set to in the interval of BTDC5degCA; in the case of a four-cylinder engine, the stroke period ΔT is the interval of 180degCA; in the case of a three-cylinder engine, the stroke period ΔT is the interval of 240degCA.

$$Q_{at}(n) = \{1 / (1 - KCCA) \times Q_{ct}(n) - KCCA / (1 - KCCA) \times Q_{ct}(n-1)\} / \Delta T(n) \quad (5)$$

<Throttle Opening Degree Control Unit **124**>

The throttle opening degree control unit **124** controls the throttle opening degree, based on the target intake air flow rate Q_{at} . Based on the target intake air flow rate Q_{at} , the throttle opening degree control unit **124** sets a target throttle opening degree TH_t and then applies driving control to the throttle motor **40** so that the real throttle opening degree TH_r approaches a target throttle opening degree TH_t . The throttle opening degree control unit **124** implements learning control for correcting the target throttle opening degree TH_t so that the real intake air flow rate Q_{ar} approaches the target intake air flow rate Q_{at} .

1-2-3. Wastegate Valve Control Unit **112**

The controller **100** is provided with the wastegate valve control unit **112**. The wastegate valve control unit **112** performs driving control of the wastegate valve **34** so as to control driving force for the compressor **31**. As represented in FIG. 2, the wastegate valve control unit **112** includes a target compressor driving force calculation unit **131** (corre-

12

sponding to a target compressor driving force calculator), a real compressor driving force calculation unit **132** (corresponding to a real compressor driving force calculator), a supercharger inertial force calculation unit **133** (corresponding to a supercharger inertial force calculator), a gate valve control value calculation unit **134** (corresponding to a gate valve control value calculator), and the like.

The target compressor driving force calculation unit **131** calculates a target compressor driving force P_{ct} , which is a target value of the driving force for the compressor **31**. The real compressor driving force calculation unit **132** calculates a real compressor driving force P_{cr} , which is a real value of the driving force for the compressor **31**. Based on the real rotation speed N_{tr} of the supercharger **36**, the supercharger inertial force calculation unit **133** calculates inertial force P_{ir} produced by an inertial moment I_t of the supercharger **36**. The gate valve control value calculation unit **134** implements driving force feedback control for changing a gate valve control value WG so that an addition value P_{cr}^* (hereinafter, referred to as inertial force added driving force P_{cr}^*) obtained by adding the inertial force P_{ir} to the real compressor driving force P_{cr} approaches the target compressor driving force P_{ct} , and performs driving control of the wastegate valve **34** based on the gate valve control value WG .

As represented in FIG. 5 in which a block diagram shows a control system in a schematic manner, when the gate valve control value calculation unit **134** (controller) changes the gate valve control value WG , the opening degree of the wastegate valve **34** changes and hence a turbine output P_t , which is driving force for the supercharger **36**, changes. In other words, when the opening degree of the wastegate valve **34** is changed, the turbine flow rate, which is the flow rate of exhaust gas that flows through the turbine **32**, and the pressure at the upstream side of the turbine **32** change; thus, the turbine output P_t to be generated by the turbine **32** changes. Then, the turbine output P_t is transmitted to the compressor **31**, where the turbine output P_t becomes driving force to be consumed in the compressor **31**. As described above, the direct control subject of the wastegate valve **34** is the exhaust system around the turbine **32** and the turbine **32**; the control system is configured in such a way that when the opening degree of the wastegate valve **34** is changed, the conditions of the exhaust system and the turbine **32** are changed and hence the turbine output P_t is changed.

Instead of directly detecting the turbine output P_t , the wastegate valve control unit **112** detects driving force actually consumed by the compressor **31**, i.e., the real compressor driving force P_{cr} . However, the turbine output P_t and the real compressor driving force P_{cr} do not completely coincide with each other. That is to say, the turbine output P_t is consumed also by inertial force P_i produced by the inertial moment I_t of the supercharger **36**. Thus, the output obtained by subtracting the inertial force P_i from the turbine output P_t is the compressor driving force P_c that is consumed by the compressor **31**.

FIG. 6 represents a comparative example in which unlike Embodiment 1, the gate valve control value WG is changed so that the detected real compressor driving force P_{cr} approaches the target compressor driving force P_{ct} . In the case of this comparative example, the inertial force P_i works as a disturbance component in the feedback system, which changes the turbine output P_t . In a transient response after the target compressor driving force P_{ct} has been changed, the rotation speed of the supercharger **36** changes; thus, the inertial force P_i becomes large. As a result, the transient response of the feedback system is disturbed by the distur-

bance component of the inertial force P_i ; thus, there have been problems, for example, that no stable transient response of the real compressor driving force P_{cr} is obtained and that the feedback control gain for obtaining a stable transient response of the real compressor driving force P_{cr} cannot appropriately be set.

Accordingly, as represented in FIG. 7, Embodiment 1 is configured in such a way that as described above, the inertial force added driving force P_{cr}^* , which is a value corresponding to the turbine output P_t , is calculated by adding the calculated inertial force P_{ir} to the detected real compressor driving force P_{cr} , and then the gate valve control value WG is changed so that the inertial force added driving force P_{cr}^* approaches the target compressor driving force P_{ct} . Thus, in the feedback control system in which the turbine output P_t is changed, the inertial force P_i , which works as a disturbance component, can be compensated and hence the feedback control can be implemented by use of the inertial force added driving force P_{cr}^* corresponding to the turbine output P_t . Accordingly, it is made possible to obtain a stable transient response of the desired inertial force added driving force P_{cr}^* for a change in the target compressor driving force P_{ct} and to appropriately set the feedback control gain for obtaining the stable transient response of the desired inertial force added driving force P_{cr}^* . Because the stable transient response of the desired inertial force added driving force P_{cr}^* can be obtained, the transient response of the real compressor driving force P_{cr} obtained by subtracting the inertial force P_{ir} from the inertial force added driving force P_{cr}^* stabilizes and hence the feedback control gain can be adjusted so that the transient response of the desired real compressor driving force P_{cr} is realized.

Hereinafter, the respective configurations of the control units in the wastegate valve control unit **112** will be explained in detail.

<Target Supercharging Pressure Calculation Unit **135**>

The wastegate valve control unit **112** is provided with a target supercharging pressure calculation unit **135** (corresponding to a target supercharging pressure calculator) for calculating a target supercharging pressure P_{2t} to be utilized in calculation of the target compressor driving force P_{ct} .

Based on the target charging efficiency E_{ct} and the real rotation speed N_{er} of the engine **1**, the target supercharging pressure calculation unit **135** calculates the target supercharging pressure P_{2t} , which is the target value of the supercharging pressure P_2 that is the pressure at a position, in the intake path **2**, that is at the downstream side of the compressor **31** and at the upstream side of the throttle valve **4**.

In Embodiment 1, based on the real rotation speed N_{er} of the engine **1** and the real manifold pressure P_{br} , the target supercharging pressure calculation unit **135** calculates the volumetric efficiency K_v on the basis of the intake manifold **5**; based on the volumetric efficiency K_v , the target charging efficiency E_{ct} , and the real intake air temperature T_{1r} , the target supercharging pressure calculation unit **135** calculates a target manifold pressure P_{bt} , which is the target value of the pressure in the intake manifold **5**; then, the target supercharging pressure calculation unit **135** adds a pressure addition value KP_2 to the target manifold pressure P_{bt} so as to calculate the target supercharging pressure P_{2t} . The volumetric efficiency K_v is a volumetric efficiency K_v on the basis of the volume of air in the intake manifold **5**, i.e., the ratio of the volume of air, in the intake manifold **5**, that is taken in by the cylinder **8**, to the cylinder volume V_c ($K_v = \frac{\text{the volume of air, in the intake manifold 5, taken in by the cylinder 8}}{V_c}$). As is the case with the real in-cylinder

fresh air amount calculation unit **142**, the target supercharging pressure calculation unit **135** calculates the volumetric efficiency K_v corresponding to the real rotation speed N_{er} and the real manifold pressure P_{br} , by use of a map in which the relationship among the rotation speed N_e , the manifold pressure P_b , and the volumetric efficiency K_v is preliminarily set.

As represented in the equation (6) below, the target supercharging pressure calculation unit **135** calculates the pressure addition value KP_2 corresponding to the target charging efficiency E_{ct} and the real rotation speed N_{er} , by use of a pressure addition value map MAP_{KP_2} in which the relationship among the rotation speed N_e , the target charging efficiency E_{ct} , and the pressure addition value KP_2 is preliminarily set. Then, the target supercharging pressure calculation unit **135** adds the pressure addition value KP_2 to the target manifold pressure P_{bt} so as to calculate the target supercharging pressure P_{2t} . The pressure addition value KP_2 is a value for securing the pressure difference between the pressure before the throttle valve **4** and the pressure after the throttle valve **4** and controlling the intake air flow rate Q_a by the throttle valve **4**. It may be allowed that the pressure addition value KP_2 is set to a fixed value of approximately 5 [kPa].

$$P_{2t} = P_{bt} + KP_2$$

$$KP_2 = MAP_{KP_2}(E_{ct}, N_{er}) \quad (6)$$

<Target Compressor Driving Force Calculation Unit **131**>

The target compressor driving force calculation unit **131** calculates the target compressor driving force P_{ct} , which is a target value of driving force for the compressor **31**.

In Embodiment 1, the target compressor driving force calculation unit **131** calculates the target compressor driving force P_{ct} , based on the target intake air flow rate Q_a and the target before/after-compressor pressure ratio P_{2t}/P_{1r} , which is the pressure ratio of the target supercharging pressure P_{2t} and the real atmospheric pressure P_{1r} .

Here, the basic characteristics of the compressor **31** and the turbine **32** will be explained. Taking the mass conservation law, the polytropic change, and the adiabatic efficiency, which are physical laws regarding the state of air, into consideration, the turbine output P_t [W] and the compressor driving force P_c [W] can be calculated through the theoretical equation (7) below.

$$P_t = Q_t \cdot \eta_t \cdot W_t = \quad (7)$$

$$Q_t \cdot \eta_t \cdot C_p \cdot T_3 \left(1 - \left(\frac{P_4}{P_3} \right)^{\frac{\kappa-1}{\kappa}} \right) = Q_t \cdot \eta_t \cdot \frac{\kappa}{\kappa-1} R \cdot T_3 \left(1 - \left(\frac{P_4}{P_3} \right)^{\frac{\kappa-1}{\kappa}} \right)$$

$$P_c = \frac{Q_{cmp} \cdot W_c}{\eta_c} = Q_{cmp} \frac{1}{\eta_c} C_p \cdot T_1 \left(\left(\frac{P_2}{P_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right) =$$

$$Q_{cmp} \frac{1}{\eta_c} \frac{\kappa}{\kappa-1} R \cdot T_1 \left(\left(\frac{P_2}{P_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right)$$

$$\therefore C_p = \frac{\kappa}{\kappa-1} R$$

where C_p is a constant-pressure specific heat [kJ/(kg·K)]; W_t is a turbine output [J] per unit flow rate; W_c is a compressor work [J] per unit flow rate; κ is a specific heat ratio; Q_t is the mass flow rate [g/s] of exhaust gas that passes through the turbine **32**; Q_{cmp} is the mass flow rate [g/s] of air that passes through the compressor **31**; R is a gas constant [kJ/(kg·K)], η_t is the adiabatic efficiency of the turbine **32**; η_c is the

adiabatic efficiency of the compressor **31**; **T3** is the temperature of exhaust gas; **P3** is the pressure at the upstream side of the turbine **32**; **P4** is the pressure at the downstream side of the turbine **32**.

Because in the normal state, the air bypass valve **33** is basically closed and hence all the intake air (the intake air flow rate Q_a) passes through the compressor **31**, it can be assumed, in the equation (7) above, that the intake air flow rate Q_a is equal to the compressor-passing flow rate Q_{cmp} . Accordingly, the compressor driving force P_c can be calculated through the equation (8) below, by use of the intake air flow rate Q_a , the before/after-compressor pressure ratio P_2/P_1 , which is the ratio of the supercharging pressure P_2 and the atmospheric pressure P_1 , and the intake-air temperature T_1 .

$$P_c = Q_a \frac{1}{\eta_c} \frac{\kappa}{\kappa - 1} R \cdot T_1 \left(\left(\frac{P_2}{P_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right) \quad (8)$$

As represented in the equation (9) below, the target compressor driving force calculation unit **131** calculates the target compressor driving force P_{ct} , based on the target intake air flow rate Q_{at} , the target before/after-compressor pressure ratio P_{2t}/P_{1r} , which is the pressure ratio of the target supercharging pressure P_{2t} and the real atmospheric pressure P_{1r} , a target adiabatic efficiency η_{ct} of the compressor **31**, and the real intake air temperature T_{1r} . In this situation, the target compressor driving force calculation unit **131** calculates a pressure ratio correction coefficient F_1 corresponding to the target before/after-compressor pressure ratio P_{2t}/P_{1r} , which is the pressure ratio of the target supercharging pressure P_{2t} and the real atmospheric pressure P_{1r} , by use of a pressure ratio correction coefficient map $MAPF_1$ in which based on the theoretical equation (9) below, the relationship between the pressure ratio correction coefficient F_1 and the before/after-compressor pressure ratio P_2/P_1 , which is the pressure ratio of the supercharging pressure P_2 to the atmospheric pressure P_1 , is preliminarily set.

$$P_{ct} = Q_{at} \frac{1}{\eta_{ct}} T_{1r} \cdot F_1 \quad (9)$$

$$F_1 = MAPF_1 \left(\frac{P_{2t}}{P_{1r}} \right),$$

$$\therefore MAPF_1 \left(\frac{P_2}{P_1} \right) \cong \frac{\kappa}{\kappa - 1} R \left(\left(\frac{P_2}{P_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right)$$

As represented in the equation (10) below, the target compressor driving force calculation unit **131** calculates the target adiabatic efficiency η_{ct} corresponding to the target intake air flow rate Q_{at} and the target before/after-compressor pressure ratio P_{2t}/P_{1r} , by use of an adiabatic efficiency calculation map $MAP\eta_c$ in which the relationship among the intake air flow rate Q_a , the before/after-compressor pressure ratio P_2/P_1 , and the adiabatic efficiency η_c of the compressor **31** is preliminarily set. It may be allowed that the target compressor driving force calculation unit **131** calculates target compressor driving force P_{ct} without considering the change in the adiabatic efficiency, for example, by setting the target adiabatic efficiency η_{ct} to a fixed value.

$$\eta_{ct} = MAP\eta_c(Q_{at}, P_{2t}/P_{1r}) \quad (10)$$

<Real Compressor Driving Force Calculation Unit **132**>

As described above, the real compressor driving force calculation unit **132** calculates the real compressor driving force P_{cr} , which is the real value of driving force for the compressor **31**.

In Embodiment 1, the target compressor driving force calculation unit **132** calculates the real compressor driving force P_{cr} , based on the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} , which is the pressure ratio of the real supercharging pressure P_{2r} and the real atmospheric pressure P_{1r} .

As represented in the after-mentioned equation (11) that is similar to the equation (9) above, the real compressor driving force calculation unit **132** calculates the real compressor driving force P_{cr} , based on the real intake air flow rate Q_{ar} , the real before/after-compressor pressure ratio P_{2r}/P_{1r} , which is the pressure ratio of the real supercharging pressure P_{2r} and the real atmospheric pressure P_{1r} , a real adiabatic efficiency η_{cr} of the compressor **31**, and the real intake air temperature T_{1r} . In this situation, as is the case with the target compressor driving force calculation unit **131**, the real compressor driving force calculation unit **132** calculates a pressure ratio correction coefficient F_1 corresponding to the real before/after-compressor pressure ratio P_{2r}/P_{1r} , which is the pressure ratio of the real supercharging pressure P_{2r} and the real atmospheric pressure P_{1r} , by use of a pressure ratio correction coefficient map $MAPF_1$ in which the relationship between the pressure ratio correction coefficient F_1 and the before/after-compressor pressure ratio P_2/P_1 is preliminarily set.

$$P_{cr} = Q_{ar} \frac{1}{\eta_{cr}} T_{1r} \cdot F_1 \quad (11)$$

$$F_1 = MAPF_1 \left(\frac{P_{2r}}{P_{1r}} \right),$$

$$\therefore MAPF_1 \left(\frac{P_2}{P_1} \right) \cong \frac{\kappa}{\kappa - 1} R \left(\left(\frac{P_2}{P_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right)$$

The real compressor driving force calculation unit **132** calculates the real adiabatic efficiency η_{cr} corresponding to the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} , by use of the adiabatic efficiency calculation map $MAP\eta_c$ in which as represented in the equation (12) below, the relationship among the intake air flow rate Q_a , the before/after-compressor pressure ratio P_2/P_1 , and the adiabatic efficiency η_c of the compressor **31** is preliminarily set. As the adiabatic efficiency calculation map $MAP\eta_c$, a map the same as the map utilized by the target compressor driving force calculation unit **131** is utilized. It may be allowed that as is the case with the target compressor driving force calculation unit **131**, the real compressor driving force calculation unit **132** calculates the real compressor driving force P_{cr} without considering the change in the adiabatic efficiency η_c , for example, by setting the real adiabatic efficiency η_{cr} to a fixed value.

$$\eta_{cr} = MAP\eta_c(Q_{ar}, P_{2r}/P_{1r}) \quad (12)$$

<Supercharger Inertial Force Calculation Unit **133**>

As described above, based on the real rotation speed N_{tr} of the supercharger **36**, the supercharger inertial force calculation unit **133** calculates inertial force P_{ir} produced by the inertial moment I_t of the supercharger **36**.

In Embodiment 1, as represented in the equation (13) below, based on the real rotation speed N_{tr} of the supercharger **36**, the supercharger inertial force calculation unit

133 calculates a real rotation acceleration α_{tr} of the supercharger 36, and then calculates the inertial force P_{ir} [W] by multiplying the real rotation acceleration α_{tr} of the supercharger 36 by the inertial moment I_t of the supercharger 36 and the real rotation speed N_{tr} of the supercharger 36. The inertial force P_{ir} [W] is power. The supercharger inertial force calculation unit 133 calculates the real rotation acceleration α_{tr} by dividing a changing amount ΔN_{tr} of the real rotation speed N_{tr} of the supercharger 36 in a preliminarily set interval ΔT_n by the interval ΔT_n .

$$P_{ir} = \alpha_{tr} \times I_t \times N_{tr}$$

$$\alpha_{tr} = \Delta N_{tr} / \Delta T_n \quad (13)$$

The inertial moment I_t is the inertial moment of the members (in this example, the turbine 32, the compressor 31, and the turbine shaft 39) that rotate integrally with the turbine 32 and the compressor 31, and is preliminarily set. <Gate Valve Control Value Calculation Unit 134>

The gate valve control value calculation unit 134 implements driving force feedback control in which the gate valve control value WG is changed so that the inertial force added driving force P_{cr}^* obtained by, as the equation (14), adding the inertial force P_{ir} to the real compressor driving force P_{cr} approaches the target compressor driving force P_{ct} . Based on the gate valve control value WG , The gate valve control value calculation unit 134 outputs a control signal to the gate valve actuator 34a so as to perform driving control of the wastegate valve 34.

$$P_{cr}^* = P_{cr} + P_{ir} \quad (14)$$

As represented in the equation (15) below, the gate valve control value calculation unit 134 implements, as driving force feedback control, PID control for calculating a feedback correction value WG_{fb} through a proportional operation, an integral operation, and a differential operation based on the difference between the target compressor driving force P_{ct} and the inertial force added driving force P_{cr}^* .

$$WG_{fb} = K_p \times \Delta P_c + \int (K_i \times \Delta P_c) dt + K_d \times d(\Delta P_c) / dt$$

$$\Delta P_c = P_{ct} - P_{cr}^* \quad (15)$$

where K_p , K_i , and K_d are the proportional gain, the integration gain, and the differentiation gain, respectively, and are control gains at a time when PID control is implemented.

It may be allowed that as the driving force feedback control, each of various kinds of feedback controls (e.g., PI control, or learning control in addition to PID control) other than PID control is utilized.

In Embodiment 1, based on the target compressor driving force P_{ct} and the exhaust gas flow rate Q_{ex} , the gate valve control value calculation unit 134 calculates a basic gate valve control value WG_b , which is the basic value of the gate valve control value WG . Then, as represented in the equation (16) below, the gate valve control value calculation unit 134 calculates, as the final gate valve control value WG , a value obtained by correcting the basic gate valve control value WG_b with the feedback correction value WG_{fb} .

$$WG = WG_b + WG_{fb} \quad (16)$$

The gate valve control value calculation unit 134 calculates, as the basic gate valve control value WG_b , the gate valve control value WG corresponding to the target compressor driving force P_{ct} and the exhaust gas flow rate Q_{ex} , by use of a gate valve control value calculation map MAP_{WG} in which as represented in the equation (17) below, the relationship among the compressor driving force

P_c , the exhaust gas flow rate Q_{ex} , and the gate valve control value WG is preliminarily set.

$$WG_b = MAP_{WG}(P_{ct}, Q_{ex}) \quad (17)$$

The gate valve control value calculation unit 134 calculates the exhaust gas flow rate Q_{ex} , based on the real intake air flow rate Q_{ar} and the air-fuel ratio AF . In Embodiment 1, as represented in the equation (18) below, the gate valve control value calculation unit 134 calculates the exhaust gas flow rate Q_{ex} , based on the real in-cylinder fresh air amount Q_{cr} calculated based on the real intake air flow rate Q_{ar} and the exhaust gas air-fuel ratio AF detected by the air-fuel ratio sensor 16. Instead of $Q_{cr} / \Delta T$, the real intake air flow rate Q_{ar} may be utilized; as the air-fuel ratio AF , the target value of the air-fuel ratio AF , which is utilized in the fuel calculation, may be utilized.

$$Q_{ex} = \frac{Q_{cr}}{\Delta T} \left(1 + \frac{1}{AF} \right) \quad (18)$$

1-2-4. Control Behavior

The behaviors of controls of the inertial force added driving force P_{cr}^* and the real compressor driving force P_{cr} will be explained by use of the timing chart in FIG. 8.

Due to an increase in the accelerator opening degree and the like, the target supercharging pressure P_{2t} increases and hence the target compressor driving force P_{ct} increases; then, the turbine output P_t and the real compressor driving force P_{cr} increase and hence the real supercharging pressure P_{2r} increases. In this situation, because the real rotation speed N_{tr} of the supercharger 36 increases, the inertial force P_{ir} of the supercharger 36 increases. Although not represented in FIG. 8, due to the driving force feedback control, the gate valve control value WG is changed in accordance with the difference between the target compressor driving force P_{ct} and the inertial force added driving force P_{cr}^* , so that the opening degree of the wastegate valve 34 is changed. Because part of the turbine output P_t is consumed as the inertial force P_{ir} , the output obtained by subtracting the inertial force P_{ir} from the turbine output P_t is the real compressor driving force P_{cr} , which is driving force consumed by the compressor 31.

In Embodiment 1, there is implemented driving force feedback control in which the gate valve control value WG is changed so that the inertial force added driving force P_{cr}^* , which is obtained by adding the inertial force P_{ir} to the real compressor driving force P_{cr} and corresponds to the turbine output P_t , approaches the target compressor driving force P_{ct} . Accordingly, as described above, in the feedback control system in which the turbine output P_t is changed, the inertial force P_{ir} , which works as a disturbance component, can be compensated and hence the feedback control can be implemented by use of the inertial force added driving force P_{cr}^* corresponding to the turbine output P_t . Accordingly, for a change in the target compressor driving force P_{ct} , a stable desired transient response of the inertial force added driving force P_{cr}^* can be obtained. In the example represented in FIG. 8, the feedback control gain is adjusted in order to obtain a feedback response in which after overshooting the target compressor driving force P_{ct} , the inertial force added driving force P_{cr}^* approaches the target compressor driving force P_{ct} . Because the real compressor driving force P_{cr} is the driving force obtained by subtracting the inertial force P_{ir} from the turbine output P_t (the inertial force added driving force P_{cr}^*), the real compressor driving force P_{cr}

becomes smaller than the inertial force added driving force P_{cr}^* that is overshooting; thus, there can be obtained a transient response in which the amount of overshooting from the target compressor driving force P_{ct} is small. Because an change in the real compressor driving force P_{cr} and an change in the real supercharging pressure P_{2r} correspond to each other, the amount of overshooting, of the real supercharging pressure P_{2r} , that is caused by a change in the target supercharging pressure P_{2t} can be reduced. That is to say, in order to obtain a transient response in which the amount of overshooting of the real compressor driving force P_{cr} from the target compressor driving force P_{ct} is small, the feedback control gain is adjusted and the transient response of the inertial force added driving force P_{cr}^* is adjusted.

FIG. 8 represents an example of case where the target supercharging pressure P_{2t} increases and hence the target compressor driving force P_{ct} increases; also in the case where the target supercharging pressure P_{2t} decreases and hence the target compressor driving force P_{ct} decreases, there can be obtained a stable transient response in which the inertial force added driving force P_{cr}^* appropriately undershoots the target compressor driving force P_{ct} . Because the real compressor driving force P_{cr} becomes larger than the inertial force added driving force P_{cr}^* that is undershooting, the undershoots of the real compressor driving force P_{cr} and the real supercharging pressure P_{2r} can be suppressed.

1-2-5. Flowchart

The procedure of the processing by the controller 100 according to Embodiment 1 will be explained based on the flowcharts in FIGS. 9 through 11. The processing items represented in the flowcharts in FIGS. 9 through 11 are recurrently implemented every constant operation cycle while the computing processing unit 90 implements software (a program) stored in the storage apparatus 91.

At first, the flowchart in FIG. 9 will be explained.

In the step S01, the driving-condition detection unit 110 implements driving condition detection processing (a driving condition detection step) for, as mentioned above, detecting the driving condition of the engine 1. In Embodiment 1, the driving-condition detection unit 110 implements supercharger rotation speed detection processing (a supercharger rotation speed detection step) for detecting the real rotation speed N_{tr} of the supercharger 36. The driving-condition detection unit 110 detects the real rotation speed N_{er} of the engine 1, the real intake air flow rate Q_{ar} , and the real atmospheric pressure P_{1r} . In addition to the foregoing driving conditions, the driving-condition detection unit 110 detects various kinds of driving conditions such as the real intake air temperature T_{1r} , the real throttle opening degree TH_r , the real manifold pressure P_{br} , the exhaust gas air-fuel ratio AF , the real supercharging pressure P_{2r} , and the accelerator opening degree D . In this situation, the driving-condition detection unit 110 (the real intake air flow rate calculation unit 141) implements real intake air flow rate calculation processing (a real intake air flow rate calculation step) for, as described above, calculating the real intake air flow rate Q_{ar} . As mentioned above, the driving-condition detection unit 110 (the real in-cylinder fresh air amount calculation unit 142) implements real in-cylinder fresh air amount calculation processing (a real in-cylinder fresh air amount calculation step) for, as described above, calculating the real charging efficiency E_{cr} and the real in-cylinder fresh air amount Q_{cr} , based on the output signal of the air flow sensor 12 or the manifold pressure sensor 15.

Next, in the step S02, the intake air control unit 111 implements intake air control processing (an intake air control step) for, as described above, controlling intake air of

the engine 1. The intake air control unit 111 calculates the target intake air flow rate Q_{at} and the target charging efficiency E_{ct} . The details of the processing in the step S02 will be represented in the flowchart in FIG. 10. In the step S10, the demanded torque calculation unit 120 implements demanded torque calculation processing (a demanded torque calculation step) for, as described above, calculating the demanded output torque TRQ_d , based on the accelerator opening degree D , a demand from an external controller, and the like. Next, in the step S11, the target torque calculation unit 121 implements target torque calculation processing (a target torque calculation step) for, as described above, calculating the target output torque TRQ_t or the target indicated mean effective pressure P_{it} , based on the demanded output torque TRQ_d . Then, in the step S12, the target in-cylinder fresh air amount calculation unit 122 implements target in-cylinder fresh air amount calculation processing (a target in-cylinder fresh air amount calculation step) for, as described above, calculating the target charging efficiency E_{ct} and the target in-cylinder fresh air amount Q_{ct} , based on the target output torque TRQ_t or the target indicated mean effective pressure P_{it} . In the step S13, the target intake air flow rate calculation unit 123 implements target intake air flow rate calculation processing (a target intake air flow rate calculation step) for, as described above, calculating the target intake air flow rate Q_{at} , based on the target in-cylinder fresh air amount Q_{ct} . In the step S14, the throttle opening degree control unit 124 implements throttle opening degree control processing (a throttle opening degree control step) for, as described above, controlling the throttle opening degree, based on the target intake air flow rate Q_{at} .

Next, in the step S03 in FIG. 9, the wastegate valve control unit 112 implements wastegate valve control processing (a wastegate valve control step) for, as described above, performing driving control of the wastegate valve 34 so as to control the supercharging pressure P_2 . The details of the processing in the step S03 will be represented in the flowchart in FIG. 11. In the step S21, the target supercharging pressure calculation unit 135 implements target supercharging pressure calculation processing (a target supercharging pressure calculation step) for, as described above, calculating the target supercharging pressure P_{2t} , based on the target charging efficiency E_{ct} and the real rotation speed N_{er} . In the step S22, the target compressor driving force calculation unit 131 implements target compressor driving force calculation processing (a target compressor driving force calculation step) for calculating the target compressor driving force P_{ct} . In Embodiment 1, as described above, the target compressor driving force calculation unit 131 calculates the target compressor driving force P_{ct} , based on the target intake air flow rate Q_{at} calculated through the intake air control step and the target before/after-compressor pressure ratio P_{2t}/P_{1r} , which is the pressure ratio of the target supercharging pressure P_{2t} and the real atmospheric pressure P_{1r} .

In the step S23, the real compressor driving force calculation unit 132 implements real compressor driving force calculation processing (a real compressor driving force calculation step) for calculating the real compressor driving force P_{cr} . In Embodiment 1, as described above, the target compressor driving force calculation unit 132 calculates the real compressor driving force P_{cr} , based on the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} , which is the pressure ratio of the real supercharging pressure P_{2r} to the real atmospheric pressure P_{1r} .

In the step S24, the supercharger inertial force calculation unit 133 implements supercharger inertial force calculation processing (a supercharger inertial force calculation step) for, as described above, calculating inertial force P_{ir} produced by the inertial moment I_t of the supercharger 36, based on the real rotation speed N_{tr} of the supercharger 36.

Then, in the step S25, the gate valve control value calculation unit 134 implements gate valve control value calculation processing (a gate valve control value calculation step) for performing driving force feedback control in which, as described above, the gate valve control value WG is changed so that the inertial force added driving force P_{cr}^* obtained by adding the inertial force P_{ir} to the real compressor driving force P_{cr} approaches the target compressor driving force P_{ct} , and performs driving control of the wastegate valve 34 based on the gate valve control value WG .

In Embodiment 1, the gate valve control value calculation unit 134 implements basic control value calculation processing for, as described above, calculating a basic value WGb of the gate valve control value WG , based on the target compressor driving force P_{ct} and the exhaust gas flow rate Q_{ex} , and then calculates, as the final gate valve control value WG , a value by correcting the basic value WGb through driving force feedback control.

2. Embodiment 2

In Embodiment 2, the supercharger inertial force calculation unit 133 calculates, as the final inertial force P_{ir} , a value obtained by applying upper/lower limitation with preliminarily set upper limit and lower limit values P_{imax} and P_{imin} to a calculation value of the inertial force P_{ir} , which is calculated based on the real rotation speed N_{tr} of the supercharger 36. The other configurations are the same as those in Embodiment 1 described above; therefore, the explanation therefor will be omitted.

The accuracy of calculating the inertial force P_{ir} is deteriorated due to an error, in detecting the real rotation speed N_{tr} of the supercharger 36, that is caused by noise components included in the output signal of the rotation speed sensor 42 for the supercharger 36 and deterioration of the accuracy of detecting the real rotation speed N_{tr} of the supercharger 36 at a time when the real rotation speed N_{tr} of the supercharger 36 is detected (estimated) based on the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} . As a result, the accuracy of calculating the inertial force added driving force P_{cr}^* that is calculated by adding the inertial force P_{ir} to the real compressor driving force P_{cr} is deteriorated; thus, the controllability of driving force feedback control may be lowered.

In Embodiment 2, as described above, the inertial force added driving force P_{cr}^* is calculated by use of the inertial force P_{ir} to which upper/lower limitation has been applied; therefore, it is made possible to prevent the inertial force P_{ir} utilized for calculation of the inertial force added driving force P_{cr}^* from excessively increasing in the positive or negative side, due to noise components in the output signal of the rotation speed sensor 42 for the supercharger 36 and the deterioration of accuracy of estimating the real rotation speed N_{tr} of the supercharger 36; as a result, the controllability of the driving force feedback control can be suppressed from being deteriorated. The upper limit value P_{imax} is set to a positive value, and the lower limit value P_{imin} is set to a negative value.

The upper/lower limitation processing by the supercharger inertial force calculation unit 133 according to Embodiment 2 can be configured as the flowchart represented in FIG. 12.

In the step S31, as is the case with Embodiment 1, the driving-condition detection unit 110 implements supercharger rotation speed detection processing (a supercharger rotation speed detection step) for detecting the real rotation speed N_{tr} of the supercharger 36. In Embodiment 2, the supercharger inertial force calculation unit 133 detects the real rotation speed N_{tr} of the supercharger 36, based on the output signal of the rotation speed sensor 42 for the supercharger 36. Alternatively, as explained in Embodiment 1, the driving-condition detection unit 110 may detect (estimate) the real rotation speed N_{tr} of the supercharger 36, based on the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} .

In the step S32, as is the case with Embodiment 1, the supercharger inertial force calculation unit 133 implements supercharger inertial force calculation processing (a supercharger inertial force calculation step) for calculating inertial force P_{ir} produced by the inertial moment I_t of the supercharger 36, based on the real rotation speed N_{tr} of the supercharger 36.

In the step S33, the supercharger inertial force calculation unit 133 determines whether or not the calculation value of the inertial force P_{ir} , which has been calculated in the step S32, is larger than the preliminarily set upper limit value P_{imax} . In the case where it is determined that the calculation value of the inertial force P_{ir} is larger than the upper limit value P_{imax} (in the step S33: Yes), the supercharger inertial force calculation unit 133 replaces the calculation value of the inertial force P_{ir} by the upper limit value P_{imax} (the step S34). In contrast, in the case where it is determined that the calculation value of the inertial force P_{ir} is not larger than the upper limit value P_{imax} (in the step S33: No), the supercharger inertial force calculation unit 133 determines whether or not the calculation value of the inertial force P_{ir} is smaller than the preliminarily set lower limit value P_{imin} (the step S35). In the case where it is determined that the calculation value of the inertial force P_{ir} is smaller than the lower limit value P_{imin} (in the step S35: Yes), the supercharger inertial force calculation unit 133 replaces the calculation value of the inertial force P_{ir} by the lower limit value P_{imin} (the step S36). In contrast, in the case where it is determined that the calculation value of the inertial force P_{ir} is not smaller than the lower limit value P_{imin} (in the step S35: No), the supercharger inertial force calculation unit 133 adopts the calculation value of the present inertial force P_{ir} , as the final inertial force P_{ir} , and then ends the processing.

3. Embodiment 3

In Embodiment 3, in the case where the calculation value P_{ir}^* calculated based on the real rotation speed N_{tr} of the supercharger 36 is within a preliminarily set falling range including zero, the supercharger inertial force calculation unit 133 adopts, as the final inertial force P_{ir} , a value decreased so as to be smaller than the calculation value P_{ir}^* . The other configurations are the same as those in Embodiment 1 described above; therefore, the explanation therefor will be omitted.

Even in the steady-driving mode where the real compressor driving force P_{cr} keeps track of the target compressor driving force P_{ct} , noise components included in the output signal of the rotation speed sensor 42 for the supercharger 36 may cause the real rotation speed N_{tr} to vibrate and hence the inertial force P_{ir} may vibrate. when the vibration of the

inertial force P_{ir} becomes large, the vibration of the inertial force added driving force P_{cr}^* to be calculated by adding the inertial force P_{ir} to the real compressor driving force P_{cr} becomes large; therefore, even in the steady-driving mode, the real compressor driving force P_{cr} may vibrate and hence the real supercharging pressure P_{2r} may vibrate. In addition, even in the case where the real rotation speed N_{tr} of the supercharger **36** is detected (estimated) based on the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} , estimation error components may cause the real rotation speed N_{tr} of the supercharger **36** to vibrate; thus, in the same manner, even in the steady-driving mode, the real compressor driving force P_{cr} may vibrate and hence the real supercharging pressure P_{2r} may vibrate.

As described above, in the case where the calculation value P_{ir}^* of the inertial force P_{ir} is within a falling range including zero, the supercharger inertial force calculation unit **133** adopts, as the final inertial force P_{ir} , a value decreased so as to be smaller than the calculation value P_{ir}^* ; therefore, the vibration of the inertial force added driving force P_{cr}^* can be reduced in the steady-driving mode. Accordingly, in the steady-driving mode, the real compressor driving force P_{cr} can be suppressed from vibrating and hence the real supercharging pressure P_{2r} can be suppressed from vibrating.

In Embodiment 3, in the case where the calculation value P_{ir}^* of the inertial force P_{ir} is within a falling range including zero, the supercharger inertial force calculation unit **133** adopts zero, as the final inertial force P_{ir} . For example, as represented in the equation (19) below, the supercharger inertial force calculation unit **133** calculates a reflection coefficient K_{pi} corresponding to the calculation value P_{ir}^* of the inertial force P_{ir} , by use of a reflection coefficient map $MAPK_{pi}$ in which the relationship, as represented in FIG. **13**, between the inertial force P_{ir} and the reflection coefficient K_{pi} is preliminarily set. In the reflection coefficient map $MAPK_{pi}$, in the case where the value of the inertial force P_{ir} is within a falling range including zero, the reflection coefficient K_{pi} is set to zero, and in the case where the value of the inertial force P_{ir} is out of the falling range, the reflection coefficient K_{pi} is set to "1". Then, as represented in the equation (19) below, the supercharger inertial force calculation unit **133** adopts, as the final inertial force P_{ir} , a value obtained by multiplying the calculation value P_{ir}^* of the inertial force P_{ir} by the reflection coefficient K_{pi} .

$$K_{pi}=MAPK_{pi}(P_{ir}^*)$$

$$P_{ir}=K_{pi}\times P_{ir}^* \quad (19)$$

The processing by the supercharger inertial force calculation unit **133** according to Embodiment 3 can be configured as the flowchart represented in FIG. **14**.

In the step **S41**, as is the case with Embodiment 1, the driving-condition detection unit **110** implements supercharger rotation speed detection processing (a supercharger rotation speed detection step) for detecting the real rotation speed N_{tr} of the supercharger **36**. In Embodiment 3, the supercharger inertial force calculation unit **133** detects the real rotation speed N_{tr} of the supercharger **36**, based on the output signal of the rotation speed sensor **42** for the supercharger **36**. Alternatively, as explained in Embodiment 1, the driving-condition detection unit **110** may detect (estimate) the real rotation speed N_{tr} of the supercharger **36**, based on the real intake air flow rate Q_{ar} and the real before/after-compressor pressure ratio P_{2r}/P_{1r} .

In the step **S42**, as is the case with Embodiment 1, the supercharger inertial force calculation unit **133** implements supercharger inertial force calculation processing (a supercharger inertial force calculation step) for calculating inertial force P_{ir}^* produced by the inertial moment I_t of the supercharger **36**, based on the real rotation speed N_{tr} of the supercharger **36**.

In the step **S43**, as described above, the supercharger inertial force calculation unit **133** implements reflection coefficient calculation processing (a reflection coefficient calculation step) for calculating the reflection coefficient K_{pi} corresponding to the calculation value P_{ir}^* of the inertial force P_{ir} , which has been calculated in the step **S42**, by use of the reflection coefficient map $MAPK_{pi}$.

Then, in the step **S43**, the supercharger inertial force calculation unit **133** calculates, as the final inertial force P_{ir} , a value obtained by multiplying the calculation value P_{ir}^* of the inertial force P_{ir} , which has been calculated in the step **S42**, by the reflection coefficient K_{pi} ; then, the supercharger inertial force calculation unit **133** ends the processing.

In the present invention, a "map" denotes a function that represents the relationship between or among a plurality of variables; instead of a map, a polynomial, a mathematical expression, a data table, or the like can be utilized.

Various modifications and alterations of this invention will be apparent to those skilled in the art without departing from the scope and spirit of this invention, and it should be understood that this is not limited to the illustrative embodiments set forth herein.

What is claimed is:

1. A controller for an internal combustion engine equipped with a supercharger having a turbine provided in an exhaust path, a compressor that is provided at the upstream side of a throttle valve in an intake path and rotates integrally with the turbine, a wastegate valve provided in a bypass, in the exhaust path, that bypasses the turbine, and a gate valve actuator that drives the wastegate valve, the controller comprising at least one processor configured to implement:

- a driving-condition detector that detects a real rotation speed of the supercharger;
- a target compressor driving force calculator that calculates a target compressor driving force, which is a target value of driving force for the compressor;
- a real compressor driving force calculator that calculates real compressor driving force, which is a real value of driving force for the compressor;
- a supercharger inertial force calculator that calculates inertial force produced by an inertial moment of the supercharger, based on the real rotation speed of the supercharger; and
- a gate valve control value calculator that implements driving force feedback control for changing a gate valve control value, which is a control value for the gate valve actuator, so that an addition value obtained by adding the inertial force to the real compressor driving force approaches the target compressor driving force, and controls the gate valve actuator to drive the wastegate valve based on the gate valve control value.

2. The controller for the internal combustion engine equipped with the supercharger according to claim **1**, wherein the supercharger inertial force calculator adopts, as a final inertial force, a value obtained by applying upper/lower limitation with preliminarily set upper limit and lower limit values to the calculation value of the inertial force, which is calculated based on the real rotation speed of the supercharger.

25

3. The controller for the internal combustion engine equipped with the supercharger according to claim 1, wherein in the case where the calculation value of the inertial force, calculated based on the real rotation speed of the supercharger, is within a preliminarily set falling range including zero, the supercharger inertial force calculator adopts, as a final inertial force, a value decreased so as to be smaller than the calculation value.

4. The controller for the internal combustion engine equipped with the supercharger according to claim 1, wherein based on the real rotation speed of the supercharger, the supercharger inertial force calculator calculates a real rotation acceleration of the supercharger, and then calculates, as the inertial force, a value obtained by multiplying the real rotation acceleration of the supercharger by the inertial moment and the real rotation speed of the supercharger.

5. The controller for the internal combustion engine equipped with the supercharger according to claim 1, wherein the driving-condition detector detects a real intake air flow rate of the internal combustion engine, a real atmospheric pressure, and a real supercharging pressure that is a real value of a supercharging pressure, which is a pressure at a position, in the intake path, that is at the downstream side of the compressor and at the upstream side of the throttle valve, and

wherein the real compressor driving force calculator calculates the real compressor driving force, based on the real intake air flow rate and a real before/after-compressor pressure ratio, which is the pressure ratio of the real supercharging pressure and the real atmospheric pressure.

6. The controller for the internal combustion engine equipped with the supercharger according to claim 1, wherein the driving-condition detector detects the real rotation speed of the supercharger, based on an output signal of a rotation speed sensor provided in the supercharger, and

wherein, alternatively, the driving-condition detector detects a real intake air flow rate of the internal combustion engine, a real atmospheric pressure, and a real supercharging pressure that is a real value of a supercharging pressure, which is a pressure at a position, in

26

the intake path, that is at the downstream side of the compressor and at the upstream side of the throttle valve, and then detects the real rotation speed of the supercharger, based on the real intake air flow rate, the real before/after-compressor pressure ratio, which is the pressure ratio of the real supercharging pressure and the real atmospheric pressure.

7. The controller for the internal combustion engine equipped with the supercharger according to claim 1, wherein the at least one processor is further configured to implement:

an intake air controller that calculates a target intake air flow rate and a target charging efficiency of the internal combustion engine; and

a target supercharging pressure calculator that calculates a target supercharging pressure, which is a target value of a supercharging pressure that is the pressure at a position, in the intake path, that is at the downstream side of the compressor and at the upstream side of the throttle valve, based on the target charging efficiency and a real rotation speed of the internal combustion engine, which is detected by the driving-condition detector,

wherein the target compressor driving force calculator calculates the target compressor driving force, based on the target intake air flow rate and a target before/after-compressor pressure ratio, which is the pressure ratio of the target supercharging pressure and the real atmospheric pressure detected by the driving-condition detector, and

wherein the gate valve control value calculator calculates an exhaust gas flow rate discharged from the internal combustion engine, based on a real intake air flow rate of the internal combustion engine, which is detected by the driving-condition detector, and an air-fuel ratio of the internal combustion engine, calculates a basic value of the gate valve control value, based on the target compressor driving force and the exhaust gas flow rate, and then calculates, as a final gate valve control value, a value obtained by correcting the basic value through the driving force feedback control.

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